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(71) Applicant: **KOMATSU LTD.**
Minato-ku, Tokyo 107 (JP)

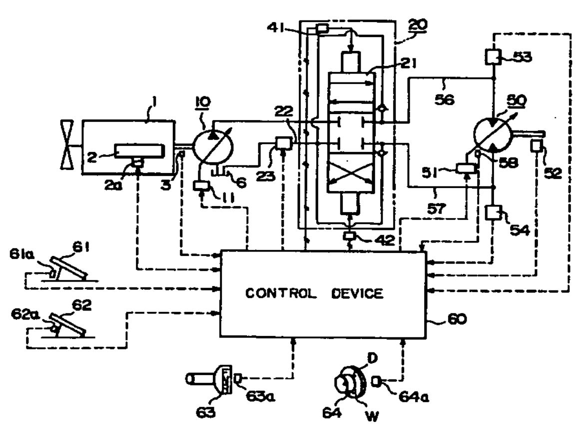
(72) Inventors:
- **OHKURA, Yasunori**
  Hiratsuka-shi Kanagawa 254 (JP)
- **HIRAKI, Hikosaburou**
  Hiratsuka-shi Kanagawa 254 (JP)
- **KANAYAMA, Noboru**
  Hiratsuka-shi Kanagawa 254 (JP)
- **MAKIDA, Ryutaro**
  Hiratsuka-shi Kanagawa 254 (JP)

(74) Representative: **Meissner, Peter E., Dipl.-Ing. et al**
Meissner & Meissner,
Patentanwaltsbüro,
Hohenzollerndamm 89
14199 Berlin (DE)

(54) **METHOD OF CONTROLLING SPEED CHANGE OF HYDRAULIC DRIVE DEVICE FOR VEHICLE AND SPEED CHANGE DEVICE**

(57) A method of controlling speed change of a hydraulic drive device for vehicles and a speed change device, which provide a favorable travelling efficiency and a favorable controllability. The method of controlling speed change comprises performing control through discriminating at least between power running and brake running from a magnitude of acceleration (θ) and a rotating speed of a hydraulic motor (50). At the time of power running, a change-over valve (21) may be fully opened in accordance with the rotating speed (ωout) of the hydraulic motor (50) and the magnitude of acceleration (θ). Also, the speed change device comprises a detecting sensor (61a) for the magnitude of acceleration, a motor rotating speed detecting sensor (52) for detecting a vehicle speed (V), and a control device (60) for discriminating between power running and brake running on the basis of the magnitude of acceleration (ωout) and the vehicle speed (V) for controlling.

FIG. 1



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Description

TECHNICAL FIELD

The present invention relates to a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device, and more particularly, to a speed change control method of a hydraulic driving apparatus for use in construction vehicles, agricultural machinery and automobiles, etc., and a speed changing device.

BACKGROUND ART

Hitherto, various types of power transmission devices for use in construction vehicles, agricultural machinery and automobiles, etc. such as of mechanical type, hydraulic type and electric type have been proposed and used. Regarding small vehicles for construction equipment, those of hydraulic type have been used relatively often. This is because those of hydraulic type can change their running speed from zero to infinity, and a merit of excellent operability has been highly regarded. On the other hand, those of hydraulic type have disadvantages of lower efficiency and higher cost to those of mechanical type. However, operating machines for digging and earth moving, etc. are mounted in construction equipment such as a wheel hydraulic excavator, and all power from an engine is converted by a hydraulic pump in order to actuate the operating machines, so that the use of hydraulic driving apparatuses may becomes less expensive conversely. When such hydraulic driving apparatuses are used, there included two types: closed circuit and open circuit. They have different characteristics, and they are used in accordance to purpose.

For example, in construction equipment used mainly for the purpose of running, when large amounts of flowing oil pressure is required, those of open circuit type are used. Closed center load sensing circuits have been used in operating machines in terms of improvement in operability, and closed center valves are adopted therein. On the other hand, when running efficiency or controllability is emphasized, those of closed circuit type are used, and a running hydraulic pump and an operating machine hydraulic pump are used therein.

In addition, a circuit using a counter balance valve shown in Fig. 37 has been known and it controls return oil from the counter balance valve to effect speed control (runaway prevention) when descending a slope. This construction comprises a variable displacement hydraulic pump 210 driven by a driving source 1 such as an engine; a capacity control device 211 for controlling capacity of the hydraulic pump 210; a forward-reverse directional control valve 212; solenoid operated proportional valve 213 (forward) and 214 (reverse) for controlling the forward-reverse directional control valve 212; a counter balance valve 215 connected to the forward-reverse directional control valve 212; a variable displacement hydraulic motor 216 for receiving the pressure oil from the counter balance valve 215; and a capacity control device 217 for controlling the capacity of the hydraulic motor 216.

However, the above-described conventional circuit encounters the following problems.

i) The use of the counter balance valve in a running circuit reduces efficiency because the valves are controlled by restriction. Moreover, since heat is generated while running, a large cooler and an engine of larger outputs are required, so that size and cost of a vehicle increases.

ii) The use of the closed center valve also in the running circuit of a vehicle emphasizing workability encounters a similar malfunction because the valve is controlled by restriction similar to the counter balance valve. Particularly, in a high-speed, long-distance running vehicle, resistance increases reducing efficiency and heating value also increases, so that a large cooler is required.

iii) The gradients of changes of the engine rotation speed are taken on the same line during low speed and during high speed, so that follow-up properties of the engine with respect to a depressing amount of the brake pedal is bad or oversensitive during either of low speed and high speed.

iv) During forward F, or reverse R, if a shock at the time of starting is decreased, or slow speed forwarding is made possible, starting properties deteriorate. If the starting properties are improved, the shock increases, or the slow speed forward cannot be effected.

v) The gradients of changes of an inclination-rotation angle of the hydraulic motor are taken on the same line during low speed and during high speed, so that the ratio of the change decreases to deteriorate follow-up properties at the time of high speed, and hunting occurs at the time of low speed.

vi) The gradients of changes of the inclination-rotation angle of the hydraulic motor are held constant with respect to a brake depressing angle, so that follow-up properties deteriorate when the brake depressing angle is large. On the other hand, when the brake depressing angle is small, the brake becomes too effective, providing greater shock.

vii) If an operating lever is operated at the time of running, the operating machine is actuated, thereby incurring dangers.

viii) The discharge capacities of the hydraulic pump are equalized during forwarding and during reversing, so that driving torque at the time of reversing for escaping from uneven ground, etc. is insufficient.

ix) A plurality of hydraulic pumps are used, each of driving apparatuses are required therefor, so that the space increases.

DISCLOSURE OF THE INVENTION

The present invention has been made to solve the drawbacks of the prior art, and its object is to provide a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device which have excellent running efficiency and controllability.

In a first aspect of a speed change control method of a hydraulic driving apparatus for vehicles according to the present invention, there is provided a speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle, the method comprising discriminating between at least a power running and a brake running from the accelerating amount and the rotation speed of the hydraulic motor for controlling.

In addition, at the time of the power running, the directional control valve may be fully opened in response to the rotation speed of the hydraulic motor and the accelerating amount, thereby reducing resistance of the directional control valve. Further, at the time of the brake running, the directional control valve may be opened at a predetermined amount in response to the rotation speed of the hydraulic motor and the accelerating amount, and the predetermined amount of opening may be increased when the accelerating amount is large. Still further, at the time of the brake running, return oil from the hydraulic motor may be allowed to have a high pressure, and fed to the supply side of the hydraulic motor when inlet pressure to the hydraulic motor is lower than permissible suction pressure of the hydraulic motor.

In a second aspect of the speed change control method, there is provided a method comprising at least one of the following: controlling absorbing torque of the hydraulic pump within a predetermined range in response to the rotation speed of the hydraulic motor and rotation speed of the engine; controlling the rotation speed of the engine within a predetermined range in response to the accelerating amount and the rotation speed of the hydraulic motor; controlling a discharge capacity of the hydraulic motor within a predetermined range in response to the rotation speed of the hydraulic motor and supply side pressure of the hydraulic motor; and controlling a discharge capacity of the hydraulic motor within a predetermined range in response to the accelerating amount and a braking amount at the time of operating a brake.

In a third aspect of the speed change control method, there is provided a method wherein, when calculating the absorbing torque of the hydraulic pump responsive to the rotation speed of the hydraulic motor and the rotation speed of the engine, the absorbing torque to be calculated takes different values when the selected position of the shifter is forward and reverse.

In a fourth aspect of the speed change control method, there is provided a method wherein, when the selected position of the shifter is forward or reverse, the directional control valve is opened at a predetermined amount in response to the accelerating amount and the rotation speed of the hydraulic motor to perform creep running.

In a fifth aspect of the speed change control method, there is provided a method comprising: selecting either a running mode or an operation mode; performing only running at the time of the running mode; and performing only operation, or both operation and running at the time of the operation mode.

In first aspect of a speed changing device a hydraulic driving apparatus for vehicles according to the present invention, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a hydraulic pump driven by the engine, a hydraulic motor for receiving pressure oil from the hydraulic pump to output a driving force and rotation speed, and a directional control valve provided between the hydraulic pump and the hydraulic motor for switching forward and reverse of the vehicle, the device comprising: an accelerating amount detection sensor for detecting the accelerating amount; a motor rotation speed detection sensor for detecting speed of the vehicle from the rotation speed of the hydraulic motor; and a control device for discriminating between a power running and a brake running by the detected accelerating amount and the detected vehicle speed for controlling.

In a second aspect of the speed changing device, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a brake pedal, and a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run a vehicle, the device comprising: a motor rotation speed sensor for detecting the speed of the vehicle from the rotation speed of the hydraulic motor; an accelerating amount detection sensor for detecting the accelerating amount; and a control device, wherein the control device, at the time of the running, fully opens the directional control valve in response to the rotation speed of the hydraulic motor and the accelerating amount to reduce resistance of the directional control valve, and/or, at the time of the brake running, opens the directional control valve at a predetermined amount in response to the rotation speed of the hydraulic motor and the accelerating amount, and increases the opening amount of the directional control valve when the accelerating amount is large.

In a third aspect of the speed changing device, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accel-

erating amount, a brake pedal, and a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run a vehicle, the device comprising: motor pressure sensors for detecting inlet pressure to the hydraulic motor and outlet pressure from the hydraulic motor; a braking amount detection sensor for detecting a braking amount of the brake pedal; a variable pressure two-stage back pressure valve for controlling return pressure of a return circuit formed between the directional control valve and an oil tank; and a control device (60), wherein the control device, at the time of the braking, compares the detected inlet pressure with permissible suction pressure of the hydraulic motor, and outputs a command to the two-stage back pressure valve to increase the return pressure when the detected inlet pressure is lower than the permissible suction pressure.

In a fourth aspect of the speed changing device, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a shifter for selecting forward and reverse of a vehicle, a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run the vehicle, and operating machine actuators for receiving the pressure oil passing through said directional control valve to drive an operating machine, the device comprising: a mode selection switch for selecting an operation mode or a running mode; a mode detection sensor for detecting the selected mode; and a control device, wherein the control device, when selecting the running mode, outputs an operation command to the directional control valve for supplying the pressure oil passing through the directional control valve to the hydraulic motor, and when selecting the operation mode, outputs either of an operation command for supplying the pressure oil passing through the directional control valve to the hydraulic motor and an operation command for supplying the pressure oil passing through the directional control valve to the hydraulic motor and the operating machine actuators.

In a fifth aspect of the speed changing device, there is provided a speed changing device of a hydraulic driving apparatus for vehicles including a driving source, an accelerator pedal for changing rotation speed of the driving source, a shifter for selecting forward and reverse of a vehicle, a hydraulic pump driven by the driving source, a hydraulic motor for receiving pressure oil from the hydraulic pump to output a driving force and rotation speed, and a closed center directional control valve provided between the hydraulic pump and the hydraulic motor for selecting forward and reverse of the vehicle in response to a selection of the shifter, the device comprising: an accelerating amount detection sensor for detecting an accelerating amount corresponding to a depressing amount of the accelerator pedal; a shifter selected position sensor for detecting a selection of the shifter; a motor rotation speed sensor for detecting rotation speed of the hydraulic motor; and a control device, wherein the control device judges whether a region is a powering region or a brake region from the accelerating amount, the change of the selected position of the shifter, and the rotation speed of the hydraulic motor.

According to such a construction of the present invention, a counter balance valve is not used in a running circuit, and the directional control valve is controlled by restriction only when it is required, so that heating while running is reduced. This eliminates the need for a large cooler and also the need for the engine of larger outputs, so that the vehicle can be reduced in size and fuel economy is improved. The cost of the vehicle is also reduced. In addition, the engine rotation speed and the discharge capacity of the hydraulic motor are changed separately during low speed and during high speed, and they are changed with a predetermined range between low speed and high speed, separate controls can be effected during low speed and during high speed, so that follow-up properties of the engine are improved. As the hydraulic motor is constructed similarly, follow-up properties with respect to the change of the required torque amount are excellent, and occurrence of hunting can be prevented even if the vehicle speed is low. Further, at the time of braking, if the brake depressing angle is large, the discharge capacity of the hydraulic motor is increased to improve follow-up properties, so that the brake becomes easily effective. On the other hand, if the brake depressing angle is small, braking can be slowed down to effect slight movement braking. In addition, the discharge capacity of the hydraulic motor while reversing is increased larger than that of while forwarding to provide large output torque, escape from uneven ground, etc. is facilitated.

Further, by selecting the operation mode, improvement in operability, which is a merit of the closed center load sensing circuit, can be obtained. On the other hand, when the running mode is selected, resistance and heating value can be reduced by opening the closed center valve, and improvement in running efficiency can be achieved. Particularly, in the vehicle having the operating machine, by using a hydraulic pump of the operating machine also in the running circuit, only one hydraulic pump is sufficient, so that space and cost of the vehicle can be reduced. In addition, as a measure to the nonuse of the counter balance valve, discharge side return oil from the hydraulic motor is sent to a supply side pipe from the hydraulic pump to the hydraulic motor through suction valves, thereby preventing occurrence of cavitation.

Further, a running mode and an operation mode are provided, and at the time of the running mode, operation can be safely performed without operation of the operating machine even if the operating lever is operated, and the operating machine takes priority at the time of the operation mode, so that the vehicle with good workability can be provided.

BRIEF DESCRIPTION OF DRAWINGS

Fig. 1 illustrates a concept of a speed changing device of a hydraulic driving apparatus for vehicles according to a first embodiment of the present invention;
Fig. 2 is a detail circuit diagram of Fig. 1;
Fig. 3 is a flowchart of an operation mode switching of the hydraulic driving apparatus according to the first embodiment;
Figs. 4 to 6 are flowcharts of a series of controls when selecting a running mode according to the first embodiment;

Figs. 7 to 18 are diagrams in the running mode according to the first embodiment;

Fig. 7 is a diagram showing the relationship among the engine rotation speed, the engine torque and the discharge amount of a hydraulic pump;
Fig. 8 is a diagram showing the relationship among the accelerating amount, the engine rotation speed, and the vehicle speed;
Fig. 9 is a diagram showing the relationship among the vehicle speed, the opening degree command value of a running valve, and the accelerating amount;
Fig. 10 is a diagram showing the relationship between frequency of a low-pass filter through which a flow rate command value is passed and the gain;
Fig. 11 is a diagram showing the relationship between the inlet pressure to a hydraulic motor and the inclined shaft command value of the hydraulic motor on running;
Fig. 12 is a diagram showing the relationship between the rotation speed of the hydraulic motor and the inclined shaft command value of the hydraulic motor on braking;
Fig. 13 is a diagram showing the relationship between frequency of a low-pass filter through which the inclined shaft command value is passed and the gain;
Fig. 14 is a diagram showing the relationship between the forward flow rate command value and the current of a forward solenoid operated directional control valve;
Fig. 15 is a diagram showing the relationship between the reverse flow rate command value and the current of a reverse solenoid operated directional control valve;
Fig. 16 is a diagram showing the relationship between the inclined shaft command value of the hydraulic motor and the current of a hydraulic motor solenoid operated directional control valve;
Fig. 17 is a diagram showing the relationship between the absorbing torque command value to a TVC valve and the current;
Fig. 18 is a diagram showing the relationship between the engine rotation speed command value and the current to a solenoid;
Figs. 19 to 21 are flowcharts of a series of controls when selecting an operation mode according to the first embodiment;

Figs. 22 to 34 are diagrams in the operation mode according to the first embodiment;

Fig. 22 is a diagram showing the relationship among the engine rotation speed, the engine torque and the discharge amount of a hydraulic pump;
Fig. 23 is a diagram showing the engine rotation speed with respect to the engine rotation setting;
Fig. 24 is a diagram showing the relationship among the accelerating amount, the accelerating correction amount, and the engine rotation speed;
Fig. 25 is a diagram showing the relationship among the vehicle speed, the opening degree command value of a running valve, and the accelerating correction amount;
Fig. 26 is a diagram showing the relationship between frequency of a low-pass filter through which the flow rate command value is passed and the gain;
Fig. 27 is a diagram showing the relationship between the inlet pressure to a hydraulic motor and the inclined shaft command value of a motor;
Fig. 28 is a diagram showing the relationship between the accelerating amount and the inclined shaft command value of a motor on braking;
Fig. 29 is a diagram showing the relationship between frequency of a low-pass filter through which the inclined shaft command value on braking is passed and the gain;
Fig. 30 is a diagram showing the relationship between the forward flow rate command value and the current of the forward solenoid operated directional control valve;
Fig. 31 is a diagram showing the relationship between the reverse flow rate command value and the current of the reverse solenoid operated directional control valve;
Fig. 32 is a diagram showing the relationship between the inclined shaft command value of the hydraulic motor on braking and the current of the hydraulic motor solenoid operated directional control valve;
Fig. 33 is a diagram showing the relationship between the absorbing torque command value to the TVC valve and the current;
Fig. 34 is a diagram showing the relationship between the fuel injection amount command value and the solenoid current;
Fig. 35 is a hydraulic circuit diagram of a speed changing device of a hydraulic driving apparatus according to a second embodiment of the present invention;

Fig. 36 is a hydraulic circuit diagram of a speed changing device of a hydraulic driving apparatus according to a third embodiment of the present invention; and

Fig. 37 is a schematic hydraulic circuit diagram showing a speed changing device of a hydraulic driving apparatus using a counter balance valve according to the prior art.

BEST MODE FOR CARRYING OUT THE INVENTION

The preferred embodiments of a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device according to the present invention will be described in detail with reference to the attached drawings.

Referring to Fig. 1, a variable displacement hydraulic pump 10 (hereinafter, referred to as a hydraulic pump 10) driven by an engine 1 sucks oil therein from an oil tank 6, converts outputs of the engine 1 into pressure oil, and feeds the pressure oil to a variable displacement hydraulic motor 50 (hereinafter, referred to as a hydraulic motor 50) via a switching device 20. The hydraulic motor 50 converts the pressure oil from the hydraulic pump 10 into rotary torque so as to drive a construction vehicle, etc. Such a hydraulic driving apparatus has already been known. To the engine 1, a fuel injection pump 2 for supplying fuels to the engine 1 is attached, and a solenoid 2a, for example, is attached to the fuel injection pump 2. The solenoid 2a receives a command from a control device 60 such as a controller, etc. to control the fuels supplied to the engine 1, and control rotation speed and outputs of the engine 1.

To the hydraulic pump 10, a servo device 11 which can allow absorbing torque to be variable is attached, and to the hydraulic motor 50, a servo device 51 which controls a swash plate, etc. for allowing displacement to be variable is attached. To the servo devices 11, 51, pressure oil is supplied via a solenoid operated directional control valve, etc. which are switched upon receipt of a command from the control device 60. The servo devices 11, 51 switch and control the pressure oil, and control absorbing torque of the hydraulic pump 10 and a discharge capacity of the hydraulic motor 50.

A running closed center load sensing valve 21 (hereinafter, referred to as a running valve 21) of the switching device 20 comprises three positions, and pilot pressures from solenoid operated directional control valves (solenoid operated proportional control valves) 41, 42, which are provided at both ends, act thereon to switch the valve 21 to a forward position or a reverse position. The solenoid operated directional control valves 41, 42 are switched upon receipt of a command from the control device 60. A return circuit 22 formed between the running valve 21 and the oil tank 6 is provided with a two-stage back pressure valve 23, which is switched in two stages by the command from the control device 60, and controls return pressure of return oil to the tank 6.

The control device 60 is provided with an engine rotation speed sensor 3 for detecting rotation speed of the engine 1, a motor rotation speed sensor 52 for detecting a rotation speed of the hydraulic motor 50 to detect a vehicle speed, and motor pressure sensors 53, 54 for detecting inlet and outlet pressures at the side of a pipe 56 or a pipe 57 which flow in and flow out of the hydraulic motor 50. To the hydraulic motor 50, an angle sensor 58 for measuring inclination-rotation angle α of a swash plate, etc. is attached to detect braking capacity of a retarder for braking a vehicle. Here, in order to make it understandable, for reasons of convenience, the motor pressure sensor 53 secured to the pipe 56 on which pressure acts in the direction of forward rotation of the hydraulic motor 50 is referred to as a forward pressure sensor 53, and the motor pressure sensor 54 secured to the opposite side pipe 57 is referred to as a reverse pressure sensor 54. Incidentally, although the motor rotation speed sensor 52 detects the rotation speed and direction, two sensors may be provided to detect the rotation speed and the direction, separately.

In addition, the control device 60 is provided with an accelerator pedal 61 for controlling a vehicle speed, a brake pedal 62 for controlling a vehicle braking, a shifter 63 for selecting a vehicle travelling direction (referred to as forward F, reverse R, and neutral N), and a mode selection switch 64. The mode selection switch 64 is the switch which actuates an operating machine switching device 45 for driving an operating machine (not shown), and selects an operation mode W for the purpose of mainly operating (including an operation while running), or a running mode D only for effecting vehicle running. The control device 60 is provided with an engine rotation setting dial 66 for setting rotation speed of the engine 1 during operation.

An accelerating amount detection sensor 61a for detecting an accelerating amount is attached to the accelerator pedal 61, a braking amount detection sensor 62a for detecting a braking amount is attached to the brake pedal 62, a shifter selected position sensor 63a for detecting a vehicle travelling direction is attached to the shifter 63, and a mode detection sensor 64a for detecting the operation mode W and the running mode D is attached to the mode selection switch 64, respectively.

Each of the above-described sensors are connected to the control device 60, and each of predetermined signals are inputted to the control device 60 therefrom.

Referring to Fig. 2, a pipe 12 is secured to a hydraulic pump 10, the pipe 12 is branched, and one pipe 12a is connected to a vehicle driving force switching device 20. The other pipe 12b is further branched into pipes 12c, 12d, 12e,.. . To the pipe 12c, an unload valve 13 is connected, and to the pipes 12d, 12e,.., a plurality of operating machine closed center load sensing valves 46a, 46b... (hereinafter, referred to as operating machine CLSS valves 46a, 46b...) are connected. The operating machine CLSS valve 46a supplies and dis-

charges pressure oil to and from operating machine actuators 47, 47a.

The servo device 11 of the hydraulic pump 10 is composed of a torque value control valve 11a (hereinafter, referred to as a TVC valve 11a), a load sensing valve 11b (hereinafter, referred to as an LS valve 11b), and a servo 11c. To the TVC valve 11a, a pilot tube 12w from the pipe 12, and the control device 60 are connected. The TVC valve 11a has a power mode function such that it receives a command from the control device 60, widely controls discharge pressure of the hydraulic pump 10 to send to the LS valve 11b, and widely and freely controls a discharge amount of the hydraulic pump 10.

For example, when a command current from the control device 60 is small, the pump discharge amount is increased to the maximum, and when the command current is large, the pump discharge amount is reduced, thereby effecting a constant horsepower control, etc. for holding pump discharge capacity x discharge pressure constant. To the LS valve 11b, a pilot tube 12w from the pipe 12 and a tube 48w of an LS circuit are connected. The LS valve 11b receives the discharge pressure PP of the hydraulic pump 10, and the highest pressure PS among pressures from the circuit (LS circuit) of the running valve 21 and operating machine CLSS valves 46a, 46b ... through check valves 49, 49a. The LS valve 11b controls pressure from the TVC valve 11a to send to the servo 11c so that an LS differential pressure (PLS = PP - PS), which is the differential pressure (PP - PS) therebetween, is held constant, thereby controlling an angle of the swash plate (the discharge amount of the pump) of the hydraulic pump 10.

The vehicle driving force switching device 20 comprises the running valve 21, a pressure compensation valve 21a, suction valves 31, 32, and safety valves 33, 34. To the running valve 21, suction valves 31, 32 and safety valves 33, 34 are attached. Each one side of the suction valve 31, 32 and the safety valve 33, 34 is connected between the running valve 21 and the hydraulic motor 50, and each other side thereof is connected between the running valve 21 and the two-stage back pressure valve 23 by means of the return circuit 22 from the running valve 21. Incidentally, the suction valves 31, 32, and the safety valves 33, 34 may be mounted to pipes without being attached to the running valve 21.

In addition, at both ends of the running valve 21, pilot pressure for switching to forward or reverse acts on predetermined both end surfaces thereof so as to switch the running valve 21 in response to an operation of the shifter 63 (Fig. 1). The pilot pressure is generated in such a manner that discharge pressure of a non-illustrated pilot pump receives a command from the control device 60, and the forward solenoid operated directional control valve 41 or the reverse solenoid operated directional control valve 42 is switched, whereby a predetermined pressure responsive to a command is generated.

A motor solenoid operated directional control valve 55 is switched upon receipt of a command from the control device 60, and pressure of the pilot pump acts on the servo device 51 so as to control displacement of the hydraulic motor 50. The hydraulic motor 50 receives the discharge amount from the hydraulic pump 10 in response to an accelerating amount so as to output a predetermined rotation speed, i.e., a predetermined vehicle speed.

The solenoid operated directional control valve 24 is switched upon receipt of a command from the control device 60, and pressure of the pilot pump acts on the two-stage back pressure valve 23 so as to control return pressure of return oil to the tank 6 in the two steps of high pressure and low pressure. When high pressure, the discharge side return oil from the hydraulic motor 50 is fed to either of the supply side pipes 56 and 57 through the suction valve 31 or 32. When low pressure, the return oil is mainly returned to the oil tank 6. At this time, a part of return oil is fed as needed to either of the pipes 56 and 57 of the supply side from the hydraulic pump 10 to the hydraulic motor 50.

Regarding an operation due to the above-described construction, a mode selection for operating a hydraulic driving apparatus will be first described.

In a non-illustrated construction vehicle having an operating machine, an operation of a hydraulic driving apparatus comprising the hydraulic pump 10, the running valve 21 and the hydraulic motor 50 is selected by a mode selection switch 64. Referring to a flow of Fig. 3, when the mode selection switch 64 is selected to a hydraulic driving control of the running mode D in step 301, a procedure advances to step 302. In step 302, the running mode D is detected by the mode detection sensor 64a, and a signal of the running mode D is sent to the control device 60. In the control device 60, a map of the running mode D and judgement are accessed. That is, a command signal of the running mode D for only effecting running is outputted from the control device 60. For example, when a non-illustrated operating machine lever is operated to operate the operating machine, the operating machine CLSS valve 46a is controlled so as not to be actuated.

In step 301, when the mode selection switch 64a is selected to a hydraulic driving control on operation (operation mode W), the procedure advances to step 303. In step 303, similar to step 302, a map of the operation mode W and judgement are accessed, and a command signal of the operation mode W mainly for the purpose of operating is outputted. For example, in performing operation while running, if the running valve 21 and the operating machine CLSS valves 46a, 46b... are operated simultaneously, a command signal mainly for the purpose of supplying to the operating machine CLSS valves 46a, 46b.. is outputted from the control device 60 when less amount is discharged from the hydraulic pump, while a command signal for supplying to the running valve 21 and the operating machine CLSS valves 46a, 46b.. is outputted from the control device 60 when large amount is discharged.

Next, a hydraulic driving control when the running

mode D is selected will be described with reference to Fig. 4, Fig. 5 and Fig. 6. Incidentally, a control of discharge capacity of the hydraulic motor 50 on running and on retarding (braking), and switching operation of the two-stage back pressure valve 23 will be described.

In step 311, various signals are inputted to the control device 60. That is, the control device 60 reads in ever-changing signals from the various sensors. The signals include a change of the shifter position (a change of forward F, reverse R, neutral N) from the shifter selected position sensor 63a, an engine rotation speed ωe from the engine rotation speed sensor 3, an accelerating amount θ from the accelerating amount detection sensor 61a, a motor rotation speed ωout (i.e., vehicle speed V of a vehicle) from the motor rotation speed sensor 52, inlet and outlet pressures Ppc from the motor pressure sensors 53, 54 to the hydraulic motor 50, and a braking capacity Rc of the retarder from the angle sensor 58. Among the inlet and outlet pressures Ppc detected by the motor pressure sensors 53, 54, inlet pressure to the hydraulic motor 50 is referred to as Pb. Incidentally, an output shaft rotation speed can be substituted for the motor rotation speed ωout.

In step 312, an acceleration Va of the vehicle is calculated from the change of the vehicle speed V, that is, determined by excluding the former ωout 1 from the ωout 2 of this time (Va = Δωout = ωout 2 - ωout 1).

In step 313, an absorbing torque command value TTVC of the hydraulic pump 10 is calculated. This is effected in such a manner that the absorbing torque is obtained by a map of Fig. 7 from the engine rotation speed ωe and the motor rotation speed ωout, and the control device 60 outputs the absorbing torque command value TTVC to the TVC valve 11a which controls discharge amount of the hydraulic pump 10.

Here, an absorbing torque control of the hydraulic pump will be described. Fig. 7 shows the horizontal axis of the engine rotation speed ωe, the vertical axis of the engine torque Te, and the absorbing torque TTVC of the hydraulic pump 10 due to the TVC valve 11a. In addition, the accelerating amount θ (θ_0 = 0, θ_1 = 1/4, θ_2 = 2/4, θ_3 = 3/4, θ_4 = 4/4 Full) is represented by slanting dotted lines. A solid line Vdh represents a change of the TVC absorbing torque when the vehicle speed is high, and a solid line Vdl represents a change of the same when the vehicle speed is low.

Referring to Fig. 2, upon receipt of the vehicle speed V from the vehicle speed sensor 52, and the engine rotation speed ωe from the engine rotation speed sensor 3, the control device 60 outputs a command to the TVC valve 11a so that the pump discharge amount x discharge pressure changes as predetermined in response to the engine rotation speed ωe.

For example, when the position of the shifter 63 is selected to neutral N at the position where the accelerating amount θ of θ_0 = 0, and at the position where the engine rotation speed ωe is on the point X1, the vehicle is suspending, or moving at slow speed by dragging torque. At this time, the hydraulic pump 10 discharges a predetermined minimum capacity, and discharged oil returns to the oil tank 6 from the unload valve 13.

When the shifter 63 is selected to forward F, the control device 60 sends a signal responsive to the accelerating amount to the forward solenoid operated directional control valve 41 by the signal from the shifter selected position sensor 63a. The solenoid operated directional control valve 41 is switched by the signal in response to the accelerating amount, controls the pilot pressure to send to the running valve 21, and opens the running valve 21 at a predetermined opening degree. When the accelerator pedal 61 is further depressed, the running valve 21 maintains a predetermined constant opening degree until a position where the accelerating amount θ is near the front of θ_1 = 1/4 (the engine rotation speed ωe is on the position of the point X2). Therefore, until the point X2, the vehicle is moving at slow speed by dragging torque in the same manner as described above, so that garaging, etc. are facilitated.

Further, when the accelerator pedal 61 is depressed, and the accelerating amount θ is on the position near the after of θ_1 = 1/4 (the engine rotation speed ωe is on the position of point X3), the running valve 21 maintains a full opening. At the same time, the control device 60 sends a command to the TVC valve 11a to actuate the servo device 11, and increases the absorbing torque (for example, displacement or pressure). This reduces a shock at the time of starting because the vehicle is started by the dragging torque at the beginning, and the speed increases after a while, and a smooth starting can be effected without any resistance because the running valve 21 is fully opened. In addition, reduction in heating value and an improvement in running efficiency can be obtained. When the engine rotation speed ωe passes the point X3, displacement of the hydraulic pump 10 increases, and the absorbing torque of the hydraulic pump 10 increases rapidly, thereby responding to the change of the vehicle speed with good follow-up properties. In addition, a large engine torque Te is taken when the vehicle speed is low for the purpose of generating a feeling such as of a torque converter.

In step 314, a fuel injection amount command value ωec to the engine 1 is calculated. That is, the engine rotation speed ωe is obtained from the accelerating amount θ and the motor rotation speed ωout shown in a map of Fig. 8, and a command signal is outputted from the control device 60 to the solenoid 2a of the fuel injection pump 2 which controls the engine rotation speed ωe.

Here, the control of the engine rotation speed ωe will be described. Referring to Fig. 8, the horizontal axis represents an accelerating amount θ, the vertical axis represents the engine rotation speed ωe, and the vehicle speed V is represented by the solid lines. The solid line Vdh represents the change of the engine rotation speed ωe when the vehicle speed is high, and the solid line Vdl represents the same when the vehicle speed is low. When the vehicle speed is high, the gradient of the

change of the engine rotation speed ωe is increased, and the gradient is decreased when the vehicle speed is low.

Upon receipt of the vehicle speed V from the vehicle speed sensor 52 and the accelerating amount θ from the accelerating amount detection sensor 61a, the control device 60 outputs a command value ωec to the solenoid 2a so that the rotation speed changes as predetermined in response to the accelerating amount θ following Fig. 8. By the command vale ωec, a predetermined injection amount is sent to the engine 1, and the engine rotation speed ωe becomes a predetermined rotation speed. For example, when the solid line Vdh of running at high speed, the maximum rotation speed is increased to absorb engine outputs as much as possible. Conversely, when low speed, motor output torque is required, so the maximum rotation speed of the engine 1 is set low. In addition, a range of an idling rotation speed is prepared, and when changing to the idling from the high speed running, the idling is set relatively high so as to increase cooling capability.

In step 315, a flow rate command value Q of the running valve 21 is calculated. This is effected in such a manner that a flow rate Q of the running valve 21 (an opening degree amount command value L of the running valve 21) is obtained by a map shown in Fig. 9 from the accelerating amount θ and the motor rotation speed ωout (the vehicle speed V of the vehicle), and the flow rate command value Q signal is outputted from the control device 60 to the solenoid operated directional control valves 41, 42 which control the flow rate Q of the running valve 21.

Here, the control of the opening degree of the running valve 21 will be descried. Referring to Fig. 9, the horizontal axis represents the vehicle speed V of the vehicle, the vertical axis represents the opening degree amount command value L of the running valve 21, and the accelerating amount θ (θ_0 = 0, θ_2 = 1/3 Full, θ_3 = 2/3 Full, θ_4 = Full) is represented by solid lines. The lower side of the slanting dotted line Lb represents a brake region LDB, and the upper side represents a running region LDD. In addition, in the running region LDD, opening of a spool shows a full opening region LDF when the opening degree command value of the running valve 21 is not less than a predetermined value La. Further, a portion of the slanting solid line of the accelerating amount θ represents a speed balance region (a CLSS control region), and a control by the running valve 21 is effected in the speed balance region. In the full opening region LDF, the opening degree of the running valve 21 is provided with predetermined openings, each becoming full in accordance with the accelerating amount θ, so that even a small accelerating amount θ is included in the full opening region LDF. This reduces a flow rate resistance, eliminates pressure collapse, and reduces heating.

Incidentally, although the opening degree amount value is shown in Fig. 9, the opening degree of the running valve 21 is proportional to the flow rate (Q = K • L) in the CLSS control. Thus, the flow rate command value Q for flowing from the hydraulic pump 10 to the hydraulic motor 50 will be obtained based on the opening degree of the running valve.

In the brake region LDB, the opening degree amount command value L of the running valve 21 is changed in response to the accelerating amount θ. In this range, the opening degree count command value L of the running valve 21 is not closed, and the larger the accelerating amount θ, the more the opening degree amount command value L is increased. Thus, even if brakes are applied to the vehicle, the vehicle does not stop suddenly, but suitably reduces speed. Further, at the left side of θ_1 shown in the drawing, similar to the above-description, a creep running is performed in which the vehicle moves at slow speed by a dragging torque. In addition, in the running mode D, the vehicle rarely runs by the control in the speed balance control between the full opening region LDF and the brake region LDB.

Referring to Fig. 9, upon receipt of the signals of the vehicle speed V and the accelerating amount θ, the control device 60 reduces the running in the speed balance control, minimizes pressure collapse due to the running valve 21, and obtains the command valve Q to be outputted to the solenoid operated directional control valves 41, 42 in accordance with Fig. 9 so as to excellently maintain brake properties.

For example, when the vehicle is running with the accelerating amount θ on the position of θ_4 = Full, and at high-speed (point X5) vehicle speed Vc, the accelerating amount θ and the vehicle speed Vc are detected by each of the sensors, and sent to the control device 60. The control device 60 receiving the signals outputs an opening degree command value Lc of the full opening region LDF for minimizing pressure collapse to the running valve 21 through the solenoid operated directional control valves 41, 42. Further, when the accelerating amount θ remains as it is, and the vehicle speed Vc increases due to a downhill, etc., the vehicle speed Vc and the opening degree command value Lc change along the line of the accelerating amount θ_4 = Full.

At this time, from point X6 to point X7 on the line of the accelerating amount θ_4 = Full, the speed balance control (CLSS control region), i.e., the opening amount of the running valve 21 gradually reduces, and a speed control for giving a braking force is effected. When the speed increases not less than the point X7, the opening amount of the running valve 21 becomes a predetermined constant value of the line Ld, and the running valve 21 gives a predetermined resistance corresponding to the speed, and generates a braking force. Over the high speed point X6, the two-stage back pressure valve 23 actuates to control the return pressure of the return oil to the tank 6 to a high pressure, and oil is supplied to the hydraulic motor 50 from either of the hydraulic pump 10 and/or suction valve 31 and the suction valve 32.

In addition, when the shifter 63 for selecting the

travelling direction of the vehicle is in forward F or reverse R, at the left side of the accelerating amount θ of θa shown in the drawing, the running valve 21 is opened by a predetermined small amount, thereby providing creep running. This creep running enables shock reduction and slow speed forwarding, as described above.

In step 316, a low-pass filter shown in Fig. 10 is multiplied by the flow rate command value Q of the running valve 21. That is, a command signal to solenoid operated directional control valves 41, 42 for switching the running valve 21 removes noises of not less than high-frequency f0 with the low-pass filter (this is expressed by Qa = Ga x Q in terms of a transfer function). This prevents very small pressure vibrations of the running valve 21, so that speed changes of the vehicle due to vibrations of the running valve 21 is eliminated.

Next, in step 317 shown in Fig. 5, whether or not the shifter 63 is operated, i.e., the position of the shifter 63 is judged by the shifter selected position sensor 63a. If neutral N, the procedure advances to step 318. If forward F, the procedure advances to step 319. If reverse R, the procedure advances to step 320.

In step 318 (in the case of neutral N position), it is checked that a forward flow rate command value QF from the control device 60 to the forward solenoid operated directional control valve 41, and a reverse flow rate command value QR to the solenoid operated directional control valve 42 are zero. When agreed with the judgement in step 317, the procedure advances to step 321.

In step 319 (in the case of forward F position), Qa is substituted into the forward flow rate command value QF, and the reverse flow rate command value QR is reduced to zero. Further, pressure Pca of the forward side pipe 56 is substituted into the inlet pressure Pp so as to be used for controlling an inclined shaft of the hydraulic motor 50. When agreed with the judgement in step 317, the procedure advances to step 321.

In step 320 (in the case of reverse R position), Qa is substituted into the reverse flow rate command value QR, and the forward flow rate command value QF is reduced to zero. Further, pressure Pcb of the reverse side pipe 57 is substituted into the inlet pressure Pp so as to be used for controlling the inclined shaft of the hydraulic motor 50 as in the case of forward F.

In step 321, whether the running region LDD or the brake region LDB is judged by a map shown in Fig. 9. If the running region LDD, the procedure advances to step 322. If the brake region LDB, the procedure advances to step 327.

In step 322, an inclined shaft command value Dd of the hydraulic motor 50 on running is calculated. The inclined shaft command value Dd controls the inclined shaft of the hydraulic motor 50 on running to control discharge capacity of the hydraulic motor 50. The motor inclined shaft control will be described with Fig. 11.

Referring to Fig. 11, the horizontal axis represents the inlet pressure Pp to the hydraulic motor 50, and the vertical axis represents the inclined shaft command value Dd of the hydraulic motor 50. The solid line Ddh shows a change of the inclined shaft command value Dd when the vehicle speed V is high, and the solid line Ddl show the same when the vehicle speed V is low. For example, when the vehicle speed is high (the solid line Ddh), the inclined shaft command value Dd is set relatively small, and the gradient of the change is increased. In addition when the vehicle speed is low (the solid line Ddl), the inclined shaft command value Dd is set relatively large, and the gradient is decreased. In addition, until the inlet pressure Pp reaches a predetermined value Ppa, the inclined shaft command value Dd maintains the minimum inclined shaft angle, and further, after the predetermined value Ppb, the inclined shaft command value Dd maintains the maximum inclined shaft angle.

In addition, based on the inlet pressure Pp detected by the motor pressure sensors 53, 54, and the vehicle speed Va detected by the vehicle speed sensor 52, the inclined shaft command value Dd is obtained from the map. The control device 60 outputs the thus obtained inclined shaft command value as a command value to the motor solenoid operated directional control valve 55 of the servo device 51.

At the time of the outputting, for example, when running at high speed of a small required torque, the inclined shaft command value Dd is decreased and the gradient of the change of the inclined shaft command value Dd is increased, thereby improving follow-up properties with respect to torque changes. In addition, when low speed of the large required torque, the inclined shaft command value Dd is increased, and the gradient of the change of the inclined shaft command value Dd is decreased to prevent hunting of the hydraulic motor 50 with respect to the torque changes. Since the inclined shaft command value Dd suitably changes in response to the required torque by providing a selection range between the high speed and the low speed, and providing a constant vehicle speed region, the percentage of change of the vehicle speed is reduced and a good response to the change of the vehicle speed is provided, thereby improving driving properties.

In step 323, changes in acceleration are examined. This judges whether or not the acceleration Va of the vehicle speed V is larger than a predetermined acceleration Vamax (a threshold value which can be arbitrarily set). If not, i.e., if smaller, the procedure advances to step 324. If larger, the procedure advances to step 328.

When the procedure advances to step 328, since the vehicle speed is increased (descending a slope, etc.) greatly, the rotation speed of the hydraulic motor 50 changes faster than a predetermined amount. For this reason, the discharge amount from the hydraulic pump 10 cannot follow up, so the control device 60 sends a signal to the two-stage back pressure valve 23, and switches the two-stage back pressure valve 23 in step 328 in the direction in which high pressure is generated. When switched to the high pressure, the discharge side return oil from the hydraulic motor 50 is fed

to the pipe 56 or the pipe 57 of the supply side from the hydraulic pump 10 through the suction valve 31 or the suction valve 32 so as to prevent occurrence of cavitation.

In step 324, entrance pressure of the hydraulic motor 50 is examined. This judges whether or not the supply side pressure Pd to the hydraulic motor 50 is lower than the threshold value of a predetermined pressure Pdmin. If not, i.e., higher than the threshold value, the procedure advances to step 325 because a non-illustrated vehicle is driven by the hydraulic motor 50. On the other hand, when lower than the threshold value, the procedure advances to step 328, and the two-stage back pressure valve 23 is switched in the direction in which high pressure is generated in the same manner as described above.

In step 325, the percentage of change of the accelerating amount θ is detected. An amount of change θa of the accelerating amount θ is obtained by the difference between the former accelerating amount θf and accelerating amount θn of this time, and whether or not the difference θa is larger than a predetermined threshold value θdec is judged. If not larger, a change of commanded speed is considered to be small, and the procedure advances to step 326. If larger, the procedure advanced to step 328.

In step 326, the two-stage back pressure valve 23 is turned off, i.e., the two-stage back pressure valve 23 is controlled to a low pressure by the command from the control device 60, whereby return oil is mainly returned to the oil tank 6. In addition, a part of return is fed to either of the supply side pipe 56 and 57 as needed through the suction valve 31 or the suction valve 32. When step 326 is completed, the procedure advances to step 329.

In step 327 to which the procedure advances in the case of the brake region LDB in the above step 321, the inclined shaft command value Dd of the hydraulic motor 50 on braking is calculated. The control of the discharge capacity of the hydraulic motor 50 on braking will be described with Fig. 12. Dotted line Dbh represents the change of the inclined shaft command value Dd when a brake depressing angle of the brake pedal 62 is high, and the solid line Dbl represents the same when the brake pedal 62 is not depressed. When the vehicle speed is high, the inclined shaft command value Dd is decreased, and the inclined shaft command value Dd is increased when the vehicle speed is low. By this, when the vehicle speed V is low, the inclined shaft command value Dd is increased, whereby a braking torque amount is increased to apply hard braking and improve follow-up properties. When the vehicle speed V is high, braking is slowed down to prevent the hard braking and maintain safety. When the discharge capacity control of the hydraulic motor 50 on braking is completed, the procedure advances to step 329 via step 328.

In step 329, a low-pass filter shown in Fig. 13 is multiplied by the inclined shaft command value Dd on braking. This allows a command signal to the solenoid operated directional control valve 55, which generates pressure for switching the hydraulic motor to pass through the low-pass filter, so as to remove high-frequency noises. This is represented by Ddc = Gb x Dd in terms of a transfer function. By the low-pass filter, very small vibrations of the hydraulic motor 50 due to high frequency is prevented, thereby eliminating speed changes of the vehicle.

In step 330, the forward flow rate command value QF is converted into a current. In addition, in step 331, the reverse flow rate command value QR is converted into a current. Referring to Fig. 14, the horizontal axis represents the forward flow rate command value QF, the vertical axis represents the current IF to the forward solenoid operated directional control valve 41. Referring to Fig. 15, the horizontal axis represents the reverse flow rate command value QR, and the vertical axis represents the current IR to the reverse solenoid operated directional control valve 42. Incidentally, in the vertical axes of Fig. 14 and later drawings, the current to the solenoid operated directional control valve is referred to as "EPC current".

As shown in Fig. 14 and Fig. 15, the forward current IF or the reverse current IR is changed with respect to the forward flow rate command value Qf or the reverse flow rate command value QR, and variations of the running valve 21 and the solenoid operated directional control valves 41, 42 are absorbed. The map is set so that the discharge capacity of the hydraulic motor 50 varies during forwarding and during reversing, and the discharge capacity during reversing is large. This increase the inclined shaft command value Dd on reversing larger than that of during forwarding to provide a large output torque, so that escape from uneven ground, etc. is facilitated.

In step 332, a current conversion command value (current Idm) is obtained by a map shown in Fig. 16 from the inclined shaft command value Dd passing through the low-pass filter, and outputted to a solenoid operated directional control valve 55. The horizontal axis represents the inclined shaft command value Dd, and the vertical axis represents the current Idm to the solenoid operated directional control valve 55. The current Idm is changed as in the case of Fig. 14 and Fig. 15 to correct non-linearity of the hydraulic motor 50 and the solenoid operated directional control valve 55.

In step 333, the command signal to the TVC valve 11a obtained in step 313 is converted into a current ITVC by a map shown in Fig. 17 to output a command. The horizontal axis represents the absorbing torque command value TTVC to the TVC valve 11a, and the vertical axis represents the current ITVC to the TVC valve 11a.

In step 334, the command signal ωec to the solenoid 2a of the fuel injection pump 2 obtained in step 314 is converted into a current Ie by a map of Fig. 18 to output a command. The horizontal axis represents the rotation speed command value ωec of engine 1, and the vertical axis represents the current Ie to the solenoid 2a.

When step 334 is completed, the procedure returns to step 311. By the flow as described above, running and brake control in the running mode D are effected.

Next, a description will be given of the usage of the vehicle by an operator in the running mode D. When only running for transportation, etc. is required, the operator. selects the running mode D by the mode selection switch 64. Then, the engine 1 is started. However, in a situation where the accelerator pedal 61 is not depressed, the engine 1 is on the position of idling and rotates at low speed, as shown in Fig. 8. When the operator operates the shifter 63 for selecting the vehicle travelling direction, and depresses the accelerator pedal 61, the vehicle will run. At this time, the control device 60 sets the rotation speed ωe of the engine 1 by Fig. 8, the running hydraulic pump 10 by Fig. 7, the opening degree amount of the running valve 21 by Fig. 9, and the inclined shaft command value Dd by Fig. 11, respectively, based on the signals from the shifter selected position sensor 63a and the accelerating amount detection sensor 61a, whereby the vehicle runs at a predetermined running speed.

By the selection of the above-described running mode D, the rotation speed ωe of the engine 1 is controlled by a depressing amount of the accelerator pedal 61. In addition, at the time of a normal running, the opening degree amount of the running valve 21 is on the position where the opening of the spool is in the full opening region LDF, and resistance at the time of running is reduced. Further, at the time of running, if the speed is increased higher than that corresponding to the accelerating amount, the opening degree amount of the running valve 21 enters the speed balance region LDD where the opening of the spool is restricted, so that the speed is controlled by the running valve 21. However, the period of time is very short, and the opening region LDF immediately enters the brake region LDB and is controlled. When entered the brake region LDB, the two-stage back pressure valve 23 is operated following the steps of the above flow, thereby preventing occurrence of cavitation.

At the time of brake operation, when the operator depresses the brake pedal 62, a signal from the braking amount detection sensor 62 enters the control device 60, and the control device 60 controls the discharge capacity of the hydraulic motor 50 in response to the depressing amount of the brake pedal 62 shown in Fig. 12 and the vehicle speed V, and applies a brake.

Next, a case will be described where the operation mode W is selected in operating mode selection.

First, an operation mode will be described. At the time of operation, if the mode selection switch 64 is switched to select the operation mode W, the operation mode W is detected by the mode detection sensor 64, and a signal of the operation mode W is sent to the control device 60. In the control device 60, a map of the operation mode W is accessed and judged. For example, only the operating machine lever is operated, the operating machine CLSS valves 46a, 46b,... are actuated, and the flow rate of each CLSS valve and the discharge amount of the hydraulic pump 10 are controlled by the highest pressure PS in the CLSS valves. In addition, when the operating machine lever is operated while running, the discharge amount of the hydraulic pump 10, and the flow rate of the operating machine CLSS valve 46a and the running valve 21 are controlled by a higher pressure which acts on the operating machine CLSS valve 46a and the running valve 21.

The hydraulic driving control when the operation mode W is selected, i.e., a case where the procedure advances to step 303 in the flow of Fig. 3 will be described with the flows of Fig. 19 to Fig. 21.

Referring to Fig. 19, in step 351, various signals are inputted to the control device 60. That is, the control device 60 reads in ever-changing signals from the various sensors. The signals, as the similar signals in step 311, include a change of the position of the shifter 63 (changes of forward F, reverse R, neutral N), the engine rotation speed ωe, the accelerating amount θ, the rotation speed ωout of the hydraulic motor 50 (the vehicle speed V), the inlet and outlet pressures Ppc, and the braking capacity Rc of the retarder. In this step, an engine rotation setting dial Erev for setting rotation of the engine 1, an operation mode selecting signal Sv for setting outputs of the operating machine, a heavy digging mode, a digging mode, and a fine operation mode, etc. according to works are further inputted.

In step 352, similar to step 312, an acceleration Vsa of the vehicle is calculated from the change of the vehicle speed V in the operation mode W. This calculation is made by excluding the former ωout 1 from the ωout 2 of this time, and is represented by the following expression:

$$Vsa = \Delta\omega out = \omega out\ 2 - \omega out\ 1$$

In step 353, an absorbing torque TSTVC of the hydraulic pump 10 in the operation mode W is calculated. This is effected in such a manner that the absorbing torque of the hydraulic pump 10 is obtained from the engine rotation speed ωe, the engine rotation setting dial Erev and the operation mode selecting signal Sv based on a map shown in Fig. 22. The obtained absorbing torque TSTVC is outputted as a command signal from the control device 60 to the TVC valve 11a.

The absorbing torque control of the hydraulic pump 10 in this case will be described. Referring to Fig. 22, the horizontal axis, the vertical axis, and the accelerating amount of dotted lines are same as those of Fig. 7. Solid lines of hyperbolas represent the absorbing torque in the operation mode W from the operation mode selecting signal Sv, high absorbing torque in a heavy cutting mode is represented by a solid line TSH, the change of the absorbing torque of the TVC when the absorbing toque is low in the fine operation mode is represented by a solid line TSL. Upon receipt of the vehicle speed V of the vehicle from the vehicle speed sensor 52, the engine rotation speed ωe from the engine rota-

tion speed sensor 3, and the operation mode selecting signal Sv, the control device 60 outputs a command to the TVC valve 11a so that the pump discharge amount x discharge pressure changes as predetermined in response to the engine rotation speed ωe.

For example, when the operator selects the heavy cutting mode by the mode selection switch 64, the operation mode selecting signal Sv of the heavy cutting mode is sent to the control device 60 from the mode detection sensor 64a, whereby the control device 60 selects the solid line TSH of high absorbing torque from a non-illustrated storage device. At this time, when the operator sets the engine rotation setting dial Erev to the position of the accelerating amount θ_4 = Full, the control device 60 selects the dotted and slanting line of the accelerating amount θ_4 = Full from the storage device. Sequentially, when a non-illustrated operating machine lever is operated, an operating machine solenoid operated directional control valve (not shown) is actuated upon receipt of a signal from the control device 60 in response to an operating amount so as to switch the operating machine CLSS valve 46a.

In accordance with the actuation, pressure from the TVC valve 11a is sent to the servo 11c so that the LS differential pressure (PLS = PP - PS) is held constant, and the inclination-rotation angle (discharge amount of the pump) of the hydraulic pump 10 is controlled. When the operating amount is increased and the discharge amount of the pump is increased, or when pressure applied to the hydraulic pump 10 is increased due to load increase of the operating machine, the absorbing torque TTVC of the hydraulic pump 10 reaches a torque curve TSH, and varies along the torque curves TSHa, TSHb, TSHc, and TSHd. That is, the absorbing torque TTVC moves sequentially on the solid lines along the points (TSHa, TSHb, TSHC, TSHd) in response to the load and the engine rotation speed ωe.

In contrast, when the operator sets the engine rotation setting dial Erev to the position of the accelerating amount θ_3 = 3/4 by the mode selection switch 64, and selects the fine operation mode, the absorbing torque TTVC sequentially moves on the solid lines along the points (TSLa, TSLb, TSLc, TSLd) in response to the load and the engine rotation speed ωe. This reduces the absorbing torque TTVC of the hydraulic pump 10 to be smaller than the absorbing toque in the heavy cutting mode, and requires less engine torque, so that reduction in fuel economy can be achieved.

. In addition, as the hydraulic pump 10, one variable displacement hydraulic pump is used. The hydraulic pump 10 sends pressure oil to actuators for driving the hydraulic driving apparatus for vehicles and an operating machine so as to drive each of them, and controls them so that the discharge amount falls within the predetermined range in response to the rotation speed of the hydraulic motor 50 and the engine. By this, in the vehicle having the operating machine, one hydraulic pump 10 is used for both operating machine and running, whereby space and cost of the vehicle can be reduced.

In step 354, a fuel injection amount command value of the engine 1 in the operation mode W is calculated. Here, the control device 60 obtains the engine rotation speed ωe from the engine rotation setting dial Erev by a map shown in Fig. 23, and outputs a command signal to the solenoid 2a which controls the engine rotation speed ωe. Referring to Fig. 23, a solid line Trev (fuel injection amount command value Trev) represents the lineal change of the engine rotation speed from the idle rotation speed to the maximum rotation speed.

For example, when the operator wishes to operate the operating machine quickly for the heavy cutting, the engine rotation setting dial Erev is set to the maximum rotation speed (full-throttle position). On the other hand, when the operator wishes to operate the operating machine slowly in the fine operation, the engine rotation setting dial Erev is set near the idle rotation speed. This allows the engine 1 to rotate corresponding to the operating amount of the engine rotation setting dial Erev. Incidentally, according to this embodiment, the control of the engine rotation speed ωe during the operation mode W, i.e., the fuel injection amount command value of the engine is set by the operating amount of the engine rotation setting dial Erev rather than the operating amount of the accelerator pedal 62.

In step 355, correction of the accelerating amount θ is obtained. This is effected in such a manner that an accelerating correction amount θs is obtained from the accelerating amount θ by Fig. 24, and a command signal is outputted from the control device 60 to the solenoid 2a of the fuel injection pump 2 which controls the engine rotation speed ωe. Fig. 24 shows the relationship between the accelerating amount θ and the accelerating correction amount θs in each engine rotation speed ωe, and a solid line NSHA represents the maximum rotation speed of the engine 1, and a solid line NSHB represents the idle rotation speed. In this embodiment, the accelerating correction amount θs is changed corresponding to the engine rotation speed ωe with respect to the same accelerating amount θ, and the accelerating correction amount θs at the idle rotation speed is smaller than the maximum rotation speed.

The correction will be described. Since the flow rate is determined by the opening amount of the running valve (closed center load sensing valve) 21 regardless of the discharge amount of the hydraulic pump 10, the opening amount is fixed and the flow rate is also fixed in the same accelerating amount θ. For this reason, if the opening amount for forwarding the vehicle at slow speed is set, the opening amount cannot be increased and the running speed is slow when a normal speed running is required. To solve this, according to this embodiment, the opening amount is increased to the normal speed, and the opening amount can be decreased when a slow speed forwarding is required. Therefore, the accelerating correction amount θs corrected with respect to the accelerating amount θ is set so as to output a small command.

In step 356, the control device 60 calculates the flow rate command value Qs of the running valve 21 (the opening degree command value Ls of the running valve 21) during the operation mode W by a map shown in Fig. 25 from the accelerating amount θ and the motor rotation speed ωout (the vehicle speed V of the vehicle), and outputs a command signal to the solenoid operated directional control valves 41, 42 for controlling the flow rate Qs. Referring to Fig. 25 which illustrates the opening degree control of the running valve 21, the horizontal axis represents the vehicle speed V of the vehicle, and the vertical axis represents the opening degree command value Ls (the flow rate command value Qs) of the running valve 21. In addition, a solid line substantially parallel to the horizontal axis represents the accelerating correction amount θs (θs_0 = 0, θs_1 = 1/4 Full, θs_2 = 2/4 Full, θs_3 = 3/4 Full, θs_4 = Full). The upper side of the dotted line Lθsa represents the running region LDD, the lower side of the dotted line Lθsb represents the brake region LDB, and a portion between the dotted line Lθsa and the dotted line Lθsb represents the speed balance region, respectively.

Referring to Fig. 25, upon receipt of the vehicle speed V from the vehicle speed sensor 52 and the accelerating amount θ from the accelerating amount detection sensor 61a, the control device 60 obtains the accelerating correction amount θs responsive to the accelerating amount θ. And, until the dotted line Lθsa where the vehicle speed reaches the predetermined vehicle speed V, the control device 60 performs the above-described CLSS control (the flow rate control responsive to the opening degree amount of the running valve 21) in response to the accelerating correction amount θ. In addition, at the predetermined vehicle speed V or more, the control device 60 decreases the opening degree command value Ls to the running valve 21 even if each of the accelerating correction amounts θs are equal. Therefore, at the vehicle speed above the dotted line Lθsa, the speed balance region is provided. In addition, even if each of the accelerating correction amounts θs are equal, the brake region LDB is provided at the vehicle speed above the dotted line Lθsb, and the brake region LDB is changed gradually in accordance with the accelerating correction amount θs.

In the above-described running region LDD, the control device 60 outputs the opening degree command value Ls from the corrected accelerating correction amount θs and the vehicle speed V to the solenoid operated directional control valve 41 or 42, and controls to effect the CLSS control. In the brake region LDB, the control device 60 changes the opening degree command value Ls (= flow rate command value Qs = Ks • Ls) in accordance with the corrected accelerating correction amount θs. In this range, the opening degree command value Ls is not closed, and the higher the speed, the more the opening degree command value Ls is increased. Thus, even if the brake is applied, the vehicle does not stop suddenly, but suitably reduces speed.

In the speed balance region, the control device 60, similar to the running region LDD, outputs the opening degree command value Ls, and controls so that the opening amount of the running valve 21 is decreased. At this control, the control device 60 outputs a command to the solenoid operated directional control valve 55 of the hydraulic motor 50 along from the parallel solid line of the accelerating correction amount θs to the solid line off to the lower left. By the command, the inclined shaft command value Dsm of the hydraulic motor 50 is varied to change the capacity of the hydraulic motor 50. Further, at θs_0 or less, similar to the above-description, a creep running is performed.

The above-described operation is, for example, performed in such a manner that the operator sets the engine rotation speed ωe to θ = 3/4 Full by the engine rotation setting dial Erev, and operates the operating lever of the operating machine and the accelerator pedal 61 to effect a simultaneous operation of the operation and the running. At the position where the accelerating correction amount θs is on the position of point Lsi of the θs_3 and the vehicle speed V = 0 with respect to the accelerating amount θ, the operating machine CLSS control is performed. The CLSS control is performed such that the operating machine CLSS valves 46a, 46b,.. are actuated by a non-illustrated operating lever, and the flow rates of each of the operating machine CLSS valves are controlled by the highest pressure PS in the CLSS valves. At this time, a discharge flow rate Qsn of the hydraulic pump 10 is the total of the operating amounts of the operating lever.

A section between the position exceeding the vehicle speed V = 0 and point X11 of the dotted line Lθsa, the operating machine is operated while running, and the flow rates of the operating machine CLSS valve 46a and the running valve 21 are controlled by the higher pressure. That is, the flow rate command value Lsi responsive to the vehicle speed V and θs_3 of the accelerating correction amount θs is outputted to the running valve 21. The discharge flow rate of the hydraulic pump 10 at this time is the total flow rate Qsn (Qsn = Qsa + Qsb + Qsc + Qsc,...) of the flow rate Qsa responsive to the accelerating correction amount θs and the vehicle speed V, and the flow rates (Qsb, Qsc, ..) responsive to the operating amounts of each of the operating levers.

A section between point X11 of the dotted line Lθsa and point X12 of the dotted line Lθsb, the operating machine is also operated while running, but this section is the speed balance region where the discharge amount from the hydraulic pump 10 is insufficient for performing simultaneous operation. In this speed balance region, supply to the operating machine takes priority. For this reason, the flow rates consistent with the opening amount are flown to the operating machine CLSS valves 46a, 46b,.., but the running valve 21 of the hydraulic motor 50 enters the speed balance region.

Here, the control device 60 outputs a command signal to the solenoid operated directional control valve 41 or 42 from the accelerating correction amount θs and the vehicle speed V, and controls the opening degree

command value Ls on line θs_3 from a parallel portion of the solid line diagonally to the lower right. In addition, the control device 60 outputs a command to the solenoid operated directional control valve 55 of the hydraulic motor 50 on line θs_3 from the parallel portion of the solid line diagonally to the lower right, and varies the inclined shaft command value Dsm to change the capacity of the hydraulic motor 50. These allow the operating machine to operate at speed consistent with the operating amount, and allow the running speed to be lower than the operating amount, whereby the simultaneous operation is effected. The brake region LDB above point X12 is similar to that described above.

In step 357, a low-pass filter of Fig. 26 is multiplied by the opening degree command value Ls (the flow rate command value Qs) of the running valve 21 during the operation mode W, whereby command signals to the solenoid operated directional control valves 41, 42 are removed therefrom high-frequency noises of not less than f0 (transfer function: Qsa = Gsa x Qs). The vertical axis represents a gain Gsa. This eliminates the speed changes of the vehicle due to vibrations of the running valve 21, similar to step 316.

In step 358 shown in Fig. 20, the position of the shifter 63 is judged. When neutral N, the procedure advances to step 359. When forward F, the procedure advances to step 360. When reverse R, the procedure advances to step 361. In the case of neutral N position, it is checked that a forward flow rate command value QSF to the forward solenoid operated directional control valve 41, and a reverse flow rate command value QSR to the reverse solenoid operated directional control valve 42 are zero. When agreed with the judgement in step 358, the procedure advances to step 362.

In step 360, Qsa is substituted into the forward flow rate command value QSF to the solenoid operated directional control valve 41, and the reverse flow rate command value QSF to the solenoid operated directional control valve 42 is reduced to zero. Further, the pressure Pca of the forward side pipe 56 is substituted into the inlet pressure Pp, and the procedure advances to step 362. In addition, in step 361, Qsa is substituted into the reverse flow rate command value QSR to the solenoid operated directional control valve 41, and the forward flow rate command value QSF to the solenoid operated directional control valve 41 is reduced to zero. Further, the pressure Pcb of the reverse side pipe 57 is substituted into the inlet pressure Pp, and the procedure advances to step 362.

In step 362, whether the running region LDD or the brake region LDB is judged. That is, in Fig. 25, if the accelerating correction amount θs and the rotation speed ωout of the hydraulic motor are above the dotted line Lθsa, it is the running region LDD, while it is the brake region LDB if they are below the dotted line Lθsb. when the running region LDD, the procedure advances to step 363 to calculate the inclined shaft command value Dsm, which controls the discharge capacity of the hydraulic motor 50 during running in the operation mode W, by a map of Fig. 27. This control is similar to that of step 322.

In step 364, similar to step 323, it is judged whether or not the acceleration Vsa of the vehicle speed V is larger than the predetermined acceleration Vamax (threshold value). In the next step 365, similar to step 324, it is judged whether or not the supply side pressure Pd to the hydraulic motor 50 during the operation mode W is lower than the predetermined pressure Pdmin.

In step 366, the amount of change θsa (the difference between the former accelerating amount θsf and the accelerating amount θsn of this time) of the accelerating correction amount θs during the operation mode W is obtained, and whether or not the amount of change θsa is larger than a predetermined threshold value θdec is judged. If larger, the procedure advances to step 369 to switch the two-stage back pressure valve 23, similar to step 328. If smaller, the procedure advances to step 367 to turn off the two-stage back pressure valve 23, and then the procedure advances to step 370. In step 370, a low-pass filter shown in Fig. 29 is multiplied by the inclined shaft command value Dsm of the hydraulic motor 50 during the operation mode W (transfer function: Dsc = Gsb x Dsm).

When the brake region LDB is judged in step 362, the procedure advances to step 368 to calculate the inclined shaft command value Dsm, which controls the discharge capacity of the hydraulic motor 50 during braking in the operation mode W, by Fig. 28. After calculating, the procedure advances to step 369 to switch the two-stage back pressure valve 23, and then the procedure advances to step 370.

Step 371 to step 375 during the operation mode W are effected in the same manner as step 330 to step 334 during the running mode D. That is, in step 371, the forward flow rate command value QSF is converted into a current Isf by a map of Fig. 30, and in step 372, the reverse flow rate command value QSR is converted into a current Isr by a map of Fig. 31. In step 373, the inclined shaft command value Dsm after the low-pass filter is converted into a current Ism by a map of Fig. 32, and in step 374, the pump absorbing torque command value TSTVC (step 353) is converted into a current IsTVC by a map of Fig. 33. In step 375, the command signal Trev (the fuel injection amount command value Trev, step 354) to the solenoid 2a is converted into a current Ise from a map of Fig. 34, and when step 375 is completed, the procedure returns to step 351.

The operation when the vehicle is used in the operation mode W will be described. When performing operation of loading or transporting cargo on board, and/or running, the operator selects the operation mode W by the mode selection switch 64. Next, when the engine 1 is actuated, and the rotation speed is set by the engine rotation setting dial 66, the engine 1 starts a constant rotation. When the vehicle is operated on a fixed position, each of actuators are controlled at the highest pressure supplied to each of the actuators by the operating lever for the operating machine, and the amount of

pressure oil supplied to each of the actuators is determined by the operating amount of the operation lever for the operating machine. Further, as the discharge amount of the hydraulic pump, the total oil amount of operating amount of each of the operating levers is discharged.

When performing running such as transportation while hoisting the operating machine, the engine 1 is the set constant rotation. At this time, when the shifter 63 for selecting the vehicle travelling direction is operated and the accelerator pedal 61 is depressed, the vehicle will run. However, the engine 1 may rotate at high speed and at middle speed. For this reason, when the rotation speed of the engine 1 is low, correction of an accelerator shown in Fig. 24 is effected so that the vehicle speed responsive to the accelerating amount becomes lower than the high rotation speed of the engine 1. At this time, the engine 1 is rotating at initially set rotation speed regardless of the accelerator pedal 61. In addition, when the discharge amount of the hydraulic pump 10 is insufficient, supply to an operating machine circuit takes priority. The discharge amount from the hydraulic pump 10 is controlled by the highest pressure in the CLSS valves for each of the operating machines and the running valve 21. In addition, the amount of pressure oil supplied to each of the actuators, and the running valve 21 are controlled as in the case when the vehicle is operated on the fixed position. The discharge amount of the hydraulic pump is, similar to the fixed position operation, the total oil amount of operating amounts of each of the operating levers for the operating machine. The operation during operating the brake is similar to that of during the running mode, but different in that a variable of the accelerating amount in the running mode is replaced with the accelerating correction amount.

Next, a second embodiment of the present invention will be described. The second embodiment differs from the first embodiment in the following point. That is, according to Fig. 2 of the first embodiment, the solenoid operated directional control valve 55 for the hydraulic motor 50 is switched by the command from the control device 60, and pressure of the pilot pump acts on the servo device 51 by this switching, and the servo device 51 controls the swash plate, etc. to allow displacement of the hydraulic motor 50 to be variable. In contrast, according to Fig. 35 showing the second embodiment, oil pressure from pipes 151, 152 of inlet and outlet ports of the hydraulic motor 50 is supplied to the servo device 51 via check valves 153, 154, and a solenoid operated directional control valve 155. This supply controls the swash plate, etc., which allows displacement of the hydraulic motor 50 to be variable, similar to the first embodiment. The solenoid operated directional control valve 155 is, similarly, switched by the command from the control device 60.

Next, an operation will be described. During a normal operation, the solenoid operated directional control valve 155 is switched to a position 155a to supply oil pressure from the pipes 151, 152 of the inlet and outlet ports of the hydraulic motor 50 to the servo device 51. By this, when the pressure acting on the hydraulic motor 50 is high, pressure to be applied to the servo device 51 is increased, and the inclined shaft angle of the hydraulic motor 50 is enlarged, thereby outputting high torque. On the other hand, when the pressure acting on the hydraulic motor 50 is low, the pressure applied to the servo device 51 is decreased and the inclined shaft angle of the hydraulic motor 50 is reduced, thereby outputting high rotation speed.

In addition, when the brake is operated, the control device 60 detects that the vehicle is in the brake region LDB shown in Fig. 9 or Fig. 25, and outputs a command to the solenoid operated directional control valve 155. This command switches the solenoid operated directional control valve 155 to a position 155b, whereby the servo device 51 is connected to the tank 6. Even if the pressure acting on the hydraulic motor 50 changes, this offers a predetermined constant pressure of the servo device 50 without any changes, so that the operation of the servo device 51, which controls the swash plate, etc. is shut down. For this reason, a braking force is fixed, and the operation is facilitated.

Then, a third embodiment of the present invention will be described with Fig. 36.

This embodiment differs from the second embodiment of Fig. 35 in the following points. That is, a swivel joint 161 is provided between the running valve 21 and the hydraulic motor 50. In addition, to the return circuit from the running valve 21 to the tank 6, a pipe 162 branched between the running valve 21 and a two-stage back pressure valve 13 is connected. The pipe 162 is connected to suction valves 131, 132 and safety valves 133, 134 via the swivel joint 161.

According to such a construction, working fluid from the safety valves 133, 134 is returned to the suction valves 131, 132 at the upstream side (the side of the suction valves) of the swivel joint 161. This eliminates occurrence of cavitation. In addition, the number of pipes in the vehicle including an upper turning body can be reduced, so that the diameter of the pipe can be enlarged, and cavitation can be effectively prevented.

INDUSTRIAL APPLICABILITY

The present invention is useful as a speed change control method of a hydraulic driving apparatus for vehicles and a speed changing device which provides good follow-up properties with respect to the engine and the torque amount change, and which can prevent hunting during low vehicle speed and provide good operability and running efficiency.

Claims

1. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes

rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle, said method comprising discriminating between at least a power running and a brake running from said accelerating amount (θ) and the rotation speed (ωout) of said hydraulic motor (50) for controlling.

2. A speed change control method of a hydraulic driving apparatus for vehicles according to claim 1, wherein, at the time of the power running, said directional control valve (21) is fully opened in response to the rotation speed (ωout) of said hydraulic motor (50) and said accelerating amount (θ), thereby reducing resistance of said directional control valve (21).

3. A speed change control method of a hydraulic driving apparatus for vehicles according to claim 1, wherein, at the time of the brake running, said directional control valve (21) is opened at a predetermined amount in response to the rotation speed (ωout) of said hydraulic motor (50) and said accelerating amount (θ), and the predetermined amount of opening of said directional control valve (21) is increased when said accelerating amount (θ) is large.

4. A speed change control method of a hydraulic driving apparatus for vehicles according to claim 1, wherein, at the time of the brake running, return oil from said hydraulic motor (50) is allowed to have a high pressure, and fed to the supply side of said hydraulic motor (50) when inlet pressure (Pp) to said hydraulic motor (50) is lower than permissible suction pressure of said hydraulic motor (50).

5. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle,

wherein said method comprising at least one of the following:

controlling absorbing torque (TTVC) of said hydraulic pump (10) within a predetermined range in response to the rotation speed (ωout) of said hydraulic motor (50) and rotation speed (ωe) of said engine (1);
controlling the rotation speed (ωe) of said engine (1) within a predetermined range in response to said accelerating amount (θ) and the rotation speed (ωout) of said hydraulic motor (50);
controlling a discharge capacity of said hydraulic motor (50) within a predetermined range in response to the rotation speed (ωout) of said hydraulic motor (50) and supply side pressure (Pd) of said hydraulic motor (50); and
controlling a discharge capacity of said hydraulic motor (50) within a predetermined range in response to said accelerating amount (θ) and a braking amount at the time of operating a brake.

6. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle,

wherein, when calculating the absorbing torque (TTVC) of said hydraulic pump (10) responsive to the rotation speed (ωout) of said hydraulic motor (50) and the rotation speed (ωe) of said engine (1), said absorbing torque (TTVC) to be calculated takes different values when the selected position of said shifter (63) is forward (F) and reverse (R).

7. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle,

wherein, when the selected position of said shifter (63) is forward (F) or reverse (R), said directional control valve (21) is opened at a predetermined amount in response to said accelerating amount (θ) and the rotation speed (ωout) of said hydraulic motor (50) to perform creep running.

8. A speed change control method of a hydraulic driving apparatus for vehicles which operates a shifter to select forward and reverse of a vehicle, changes rotation speed of an engine by an accelerating amount, and supplies pressure oil from a hydraulic pump driven by the engine to a hydraulic motor through a directional control valve so as to control rotation speed of the hydraulic motor for running the vehicle, said method comprising:

selecting either a running mode (D) and an

operation mode (W);
performing only running at the time of said running mode (D); and
performing only operation, or both of operation and running at the time of said operation mode (W).

9. A speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a hydraulic pump driven by the engine, a hydraulic motor for receiving pressure oil from the hydraulic pump to output a driving force and rotation speed, and a directional control valve provided between the hydraulic pump and the hydraulic motor for switching forward and reverse of the vehicle, said device comprising:

an accelerating amount detection sensor (61a) for detecting said accelerating amount (θ); a motor rotation speed detection sensor (52) for detecting speed (V) of said vehicle from the rotation speed (ωout) of said hydraulic motor (50); and a control device (60) for discriminating between a power running and a brake running by said detected accelerating amount (θ) and said detected vehicle speed (V) for controlling.

10. A speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a brake pedal, and a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run a vehicle, said device comprising:

a motor rotation speed sensor (52) for detecting the speed (V) of said vehicle from the rotation speed (ωout) of said hydraulic motor (50); an accelerating amount detection sensor (61a) for detecting said accelerating amount (θ); and a control device (60),
wherein said control device (60), at the time of the running, fully opens said directional control valve (21) in response to the rotation speed (ωout) of said hydraulic motor (50) and said accelerating amount (θ) to reduce resistance of said directional control valve (21), and/or, at the time of the brake running, opens said directional control valve (21) at a predetermined amount in response to the rotation speed (ωout) of said hydraulic motor (50) and said accelerating amount (θ), and increases the opening amount of said directional control valve (21) when said accelerating amount (θ) is large.

11. A speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a brake pedal, and a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run a vehicle, said device comprising:

motor pressure sensors (53, 54) for detecting inlet pressure (Pp) to said hydraulic motor (50) and outlet pressure from said hydraulic motor (50); a braking amount detection sensor (62a) for detecting a braking amount of said brake pedal (62); a variable pressure two-stage back pressure valve (23) for controlling return pressure of a return circuit (22) formed between said directional control valve (21) and an oil tank (6); and a control device (60),
wherein said control device (60), at the time of the braking, compares said detected inlet pressure (Pp) with permissible suction pressure of said hydraulic motor (50), and outputs a command to said two-stage back pressure valve (23) to increase said return pressure when said detected inlet pressure (Pp) is lower than the permissible suction pressure.

12. A speed changing device of a hydraulic driving apparatus for vehicles including an engine of which rotation speed changes in response to an accelerating amount, a shifter for selecting forward and reverse of a vehicle, a hydraulic motor for receiving pressure oil from a hydraulic pump through a directional control valve to run the vehicle, and operating machine actuators for receiving the pressure oil passing through said directional control valve to drive an operating machine, said device comprising:

a mode selection switch (64) for selecting an operation mode (W) or a running mode (D); a mode detection sensor (64a) for detecting said selected mode; and a control device (60),
wherein said control device (60), when selecting said running mode (D), outputs an operation command to said directional control valve (21) for supplying the pressure oil passing through said directional control valve (21) to said hydraulic motor, and when selecting said operation mode (W), outputs either of an operation command for supplying the pressure oil passing through said directional control valve (21) to said hydraulic motor (50) and an operation command for supplying the pressure oil passing through said directional control valve (21) to said hydraulic motor (50) and said operating machine actuators.

13. A speed changing device of a hydraulic driving apparatus for vehicles including a driving source,

an accelerator pedal for changing rotation speed of the driving source, a shifter for selecting forward and reverse of a vehicle, a hydraulic pump driven by the driving source, a hydraulic motor for receiving pressure oil from the hydraulic pump to output a driving force and rotation speed, and a closed center directional control valve provided between the hydraulic pump and the hydraulic motor for selecting forward and reverse of the vehicle in response to a selection of the shifter, said device comprising:

an accelerating amount detection sensor (61a) for detecting an accelerating amount (θ) corresponding to a depressing amount of said accelerator pedal (61); a shifter selected position sensor (63a) for detecting a selection of said shifter (63); a motor rotation speed sensor (52) for detecting rotation speed (ωout) of said hydraulic motor (50); and a control device (60),

wherein said control device (60) judges whether a region is a powering region (LDD) or a brake region (LDB) from said accelerating amount (θ), the change of the selected position of said shifter (63), and the rotation speed (ωout) of said hydraulic motor (50).

FIG. 1

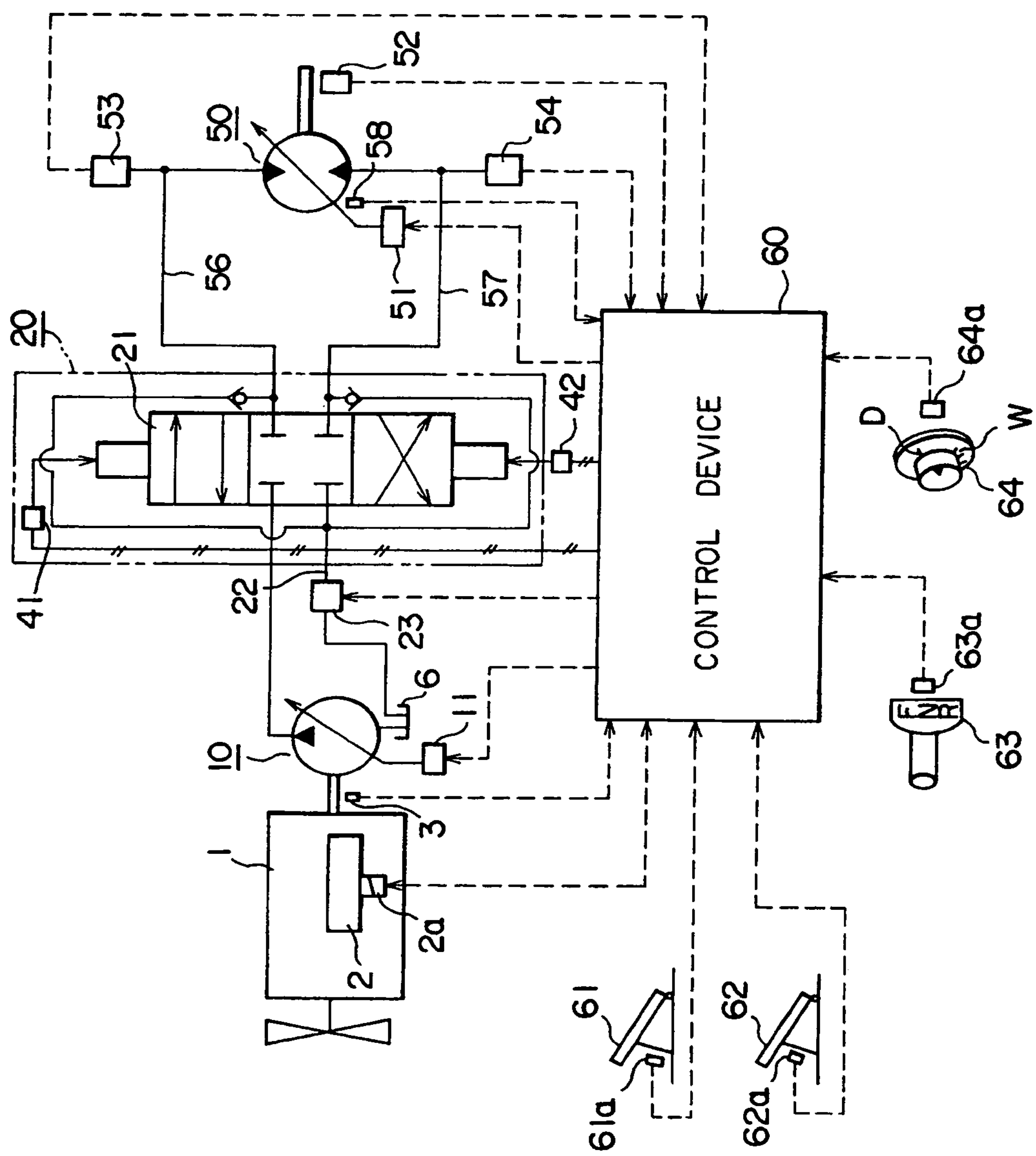

CONTROL DEVICE
F N R
D
W
1
2
2a
3
6
10
11
20
21
22
23
41
42
50
51
52
53
54
56
57
58
60
61
61a
62
62a
63
63a
64
64a

FIG. 2

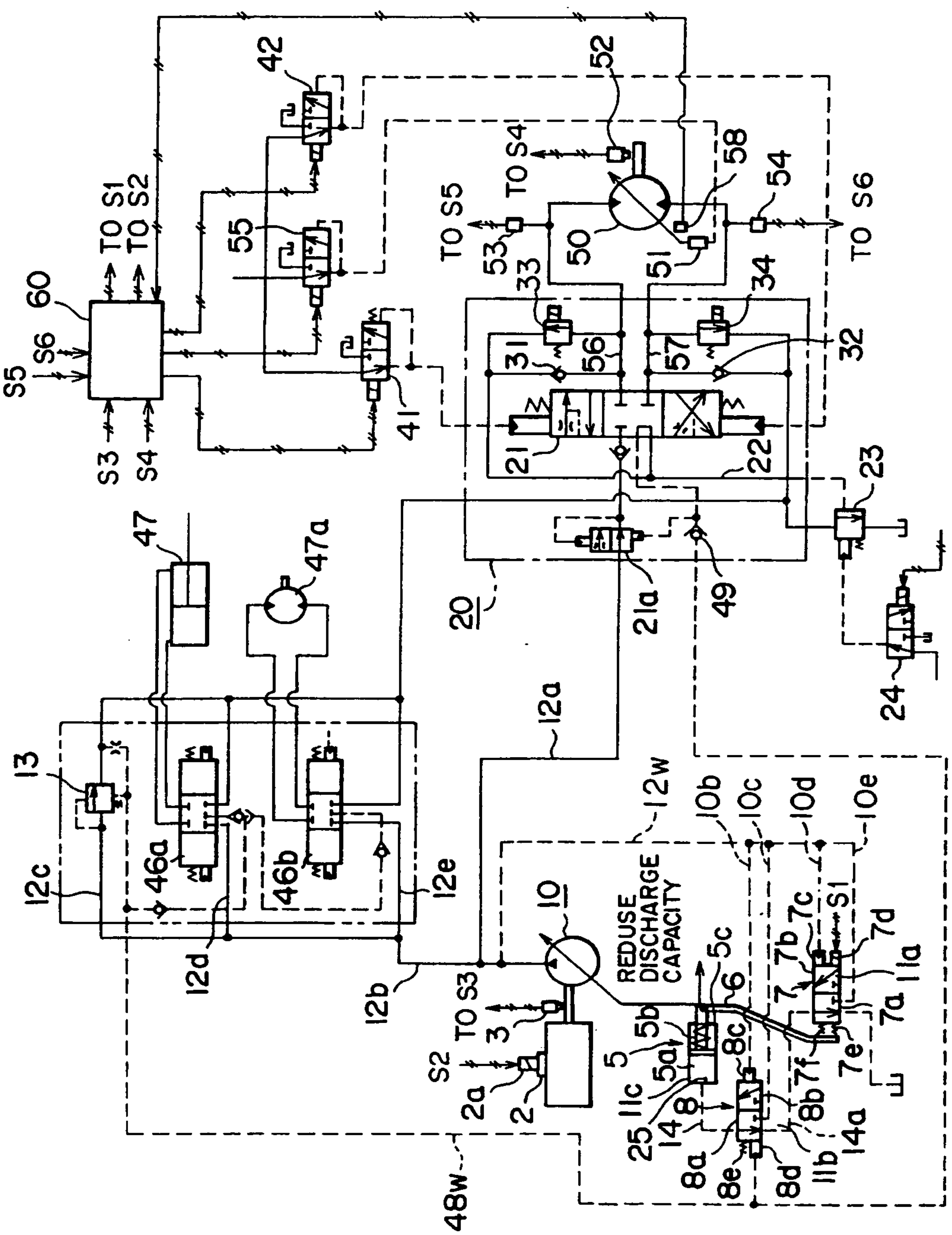


FIG. 3

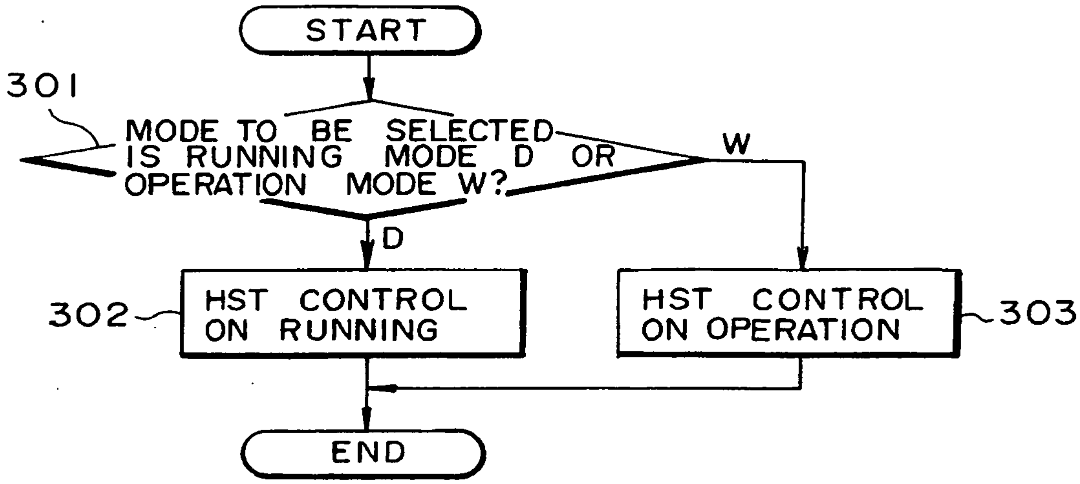

START
301
MODE TO BE SELECTED IS RUNNING MODE D OR OPERATION MODE W?
W
D
302
HST CONTROL ON RUNNING
HST CONTROL ON OPERATION
303
END

FIG. 4

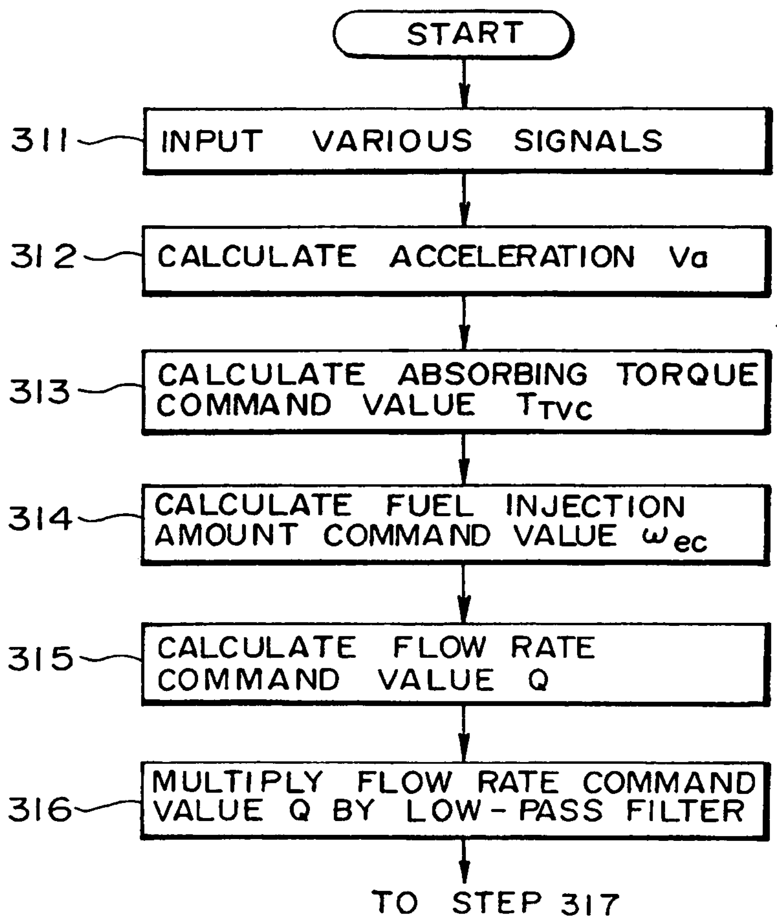

START
311
INPUT VARIOUS SIGNALS
312
CALCULATE ACCELERATION Va
313
CALCULATE ABSORBING TORQUE COMMAND VALUE TTVC
314
CALCULATE FUEL INJECTION AMOUNT COMMAND VALUE ωec
315
CALCULATE FLOW RATE COMMAND VALUE Q
316
MULTIPLY FLOW RATE COMMAND VALUE Q BY LOW-PASS FILTER
TO STEP 317

FIG. 5

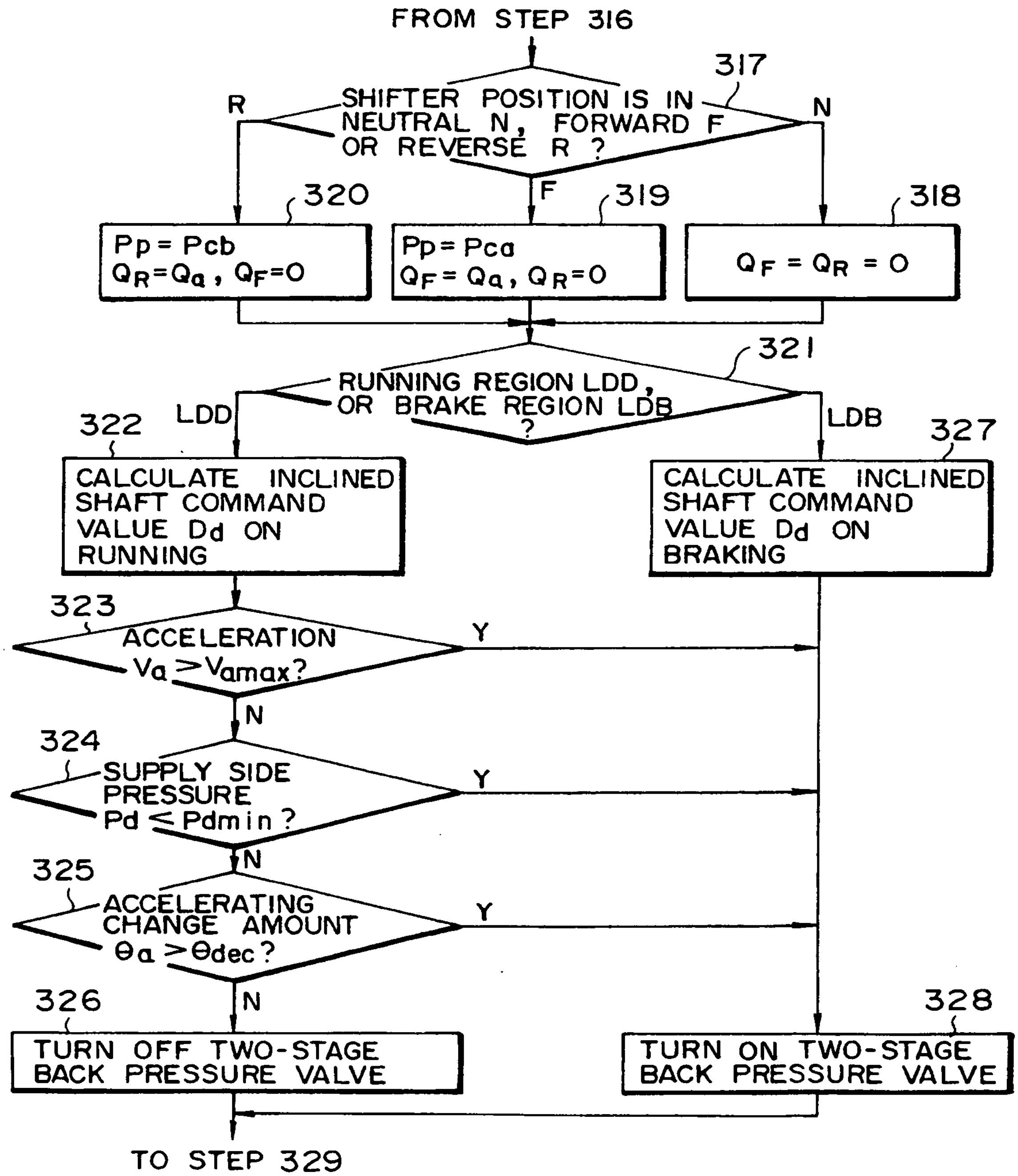

FROM STEP 316
317
SHIFTER POSITION IS IN NEUTRAL N, FORWARD F OR REVERSE R ?
R
N
F
320
Pp = Pcb
QR=Qa , QF=0
319
Pp = Pca
QF = Qa , QR=0
318
QF = QR = 0
321
RUNNING REGION LDD, OR BRAKE REGION LDB ?
LDD
LDB
322
CALCULATE INCLINED SHAFT COMMAND VALUE Dd ON RUNNING
327
CALCULATE INCLINED SHAFT COMMAND VALUE Dd ON BRAKING
323
ACCELERATION Va > Vamax?
Y
N
324
SUPPLY SIDE PRESSURE Pd ≤ Pdmin ?
Y
N
325
ACCELERATING CHANGE AMOUNT θa > θdec ?
Y
N
326
TURN OFF TWO-STAGE BACK PRESSURE VALVE
328
TURN ON TWO-STAGE BACK PRESSURE VALVE
TO STEP 329

FIG. 6

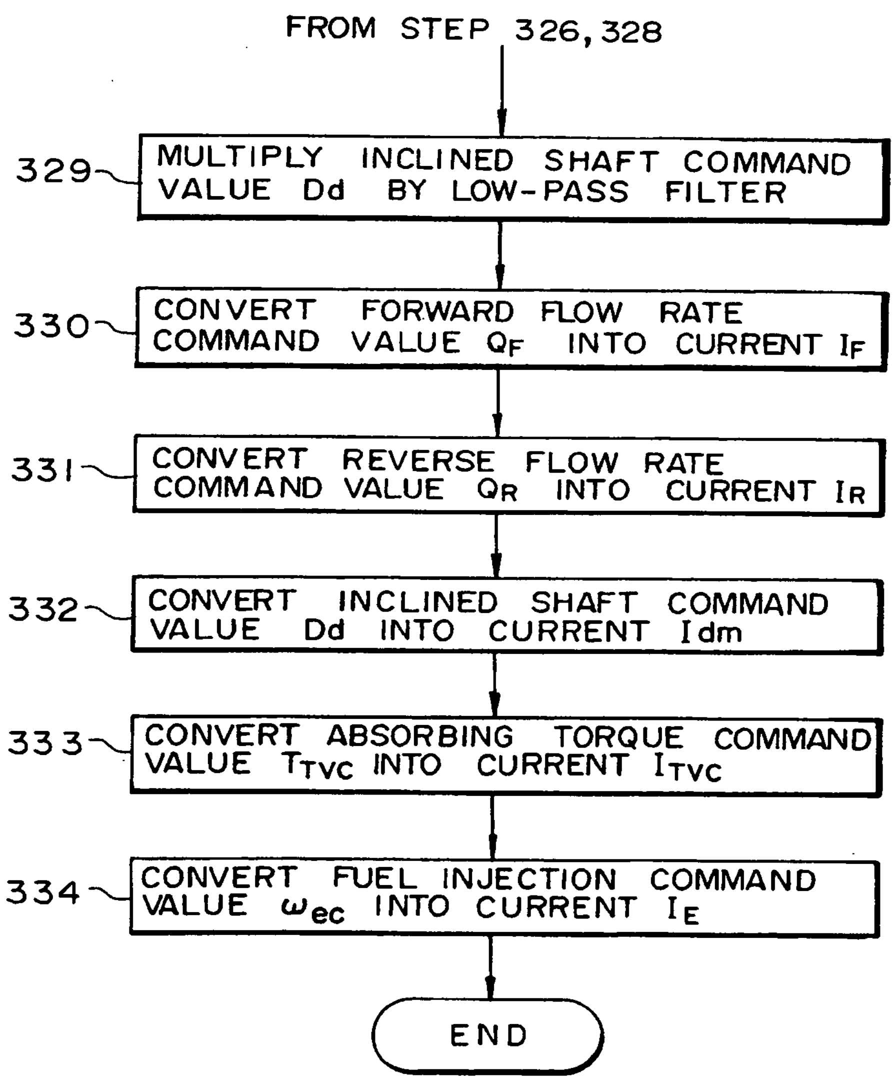

FROM STEP 326, 328
329
MULTIPLY INCLINED SHAFT COMMAND VALUE Dd BY LOW-PASS FILTER
330
CONVERT FORWARD FLOW RATE COMMAND VALUE Q_F INTO CURRENT I_F
331
CONVERT REVERSE FLOW RATE COMMAND VALUE Q_R INTO CURRENT I_R
332
CONVERT INCLINED SHAFT COMMAND VALUE Dd INTO CURRENT Idm
333
CONVERT ABSORBING TORQUE COMMAND VALUE T_{TVC} INTO CURRENT I_{TVC}
334
CONVERT FUEL INJECTION COMMAND VALUE ω_{ec} INTO CURRENT I_E
END

FIG. 7

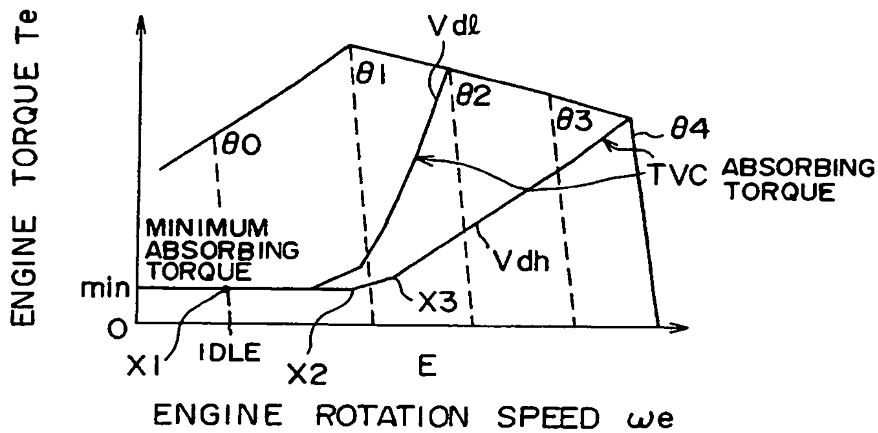


FIG. 8

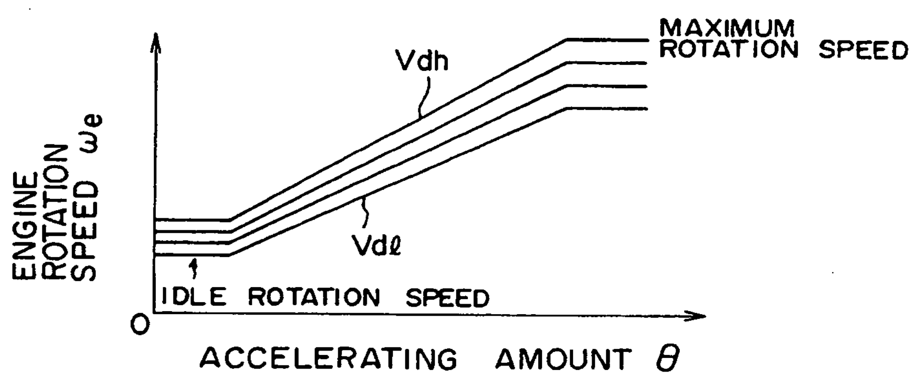


FIG. 9

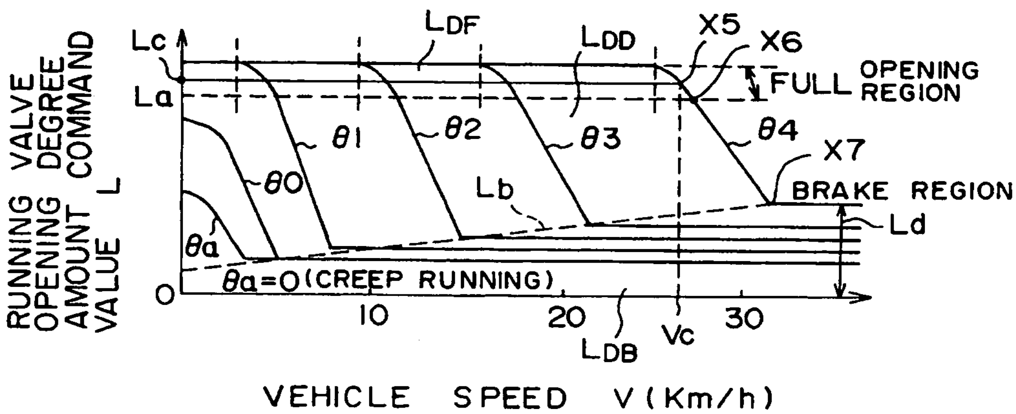


FIG. 10

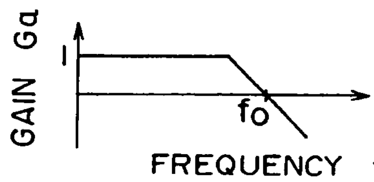


FIG. 11

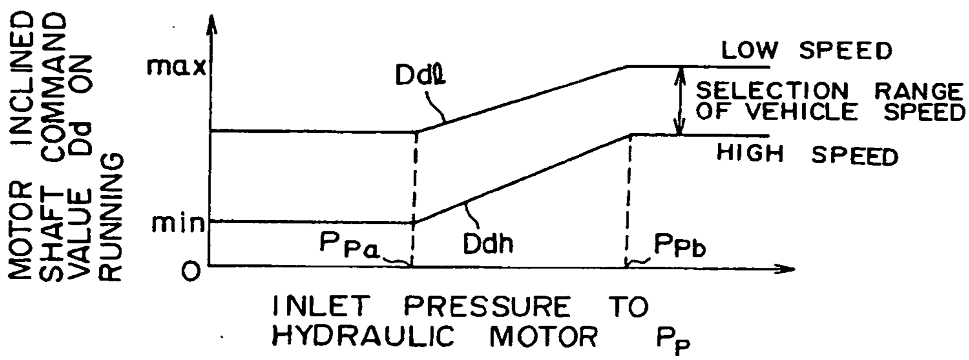


FIG. 12

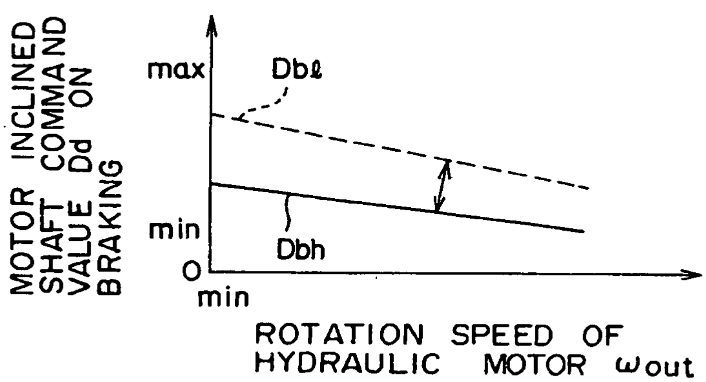


FIG. 13

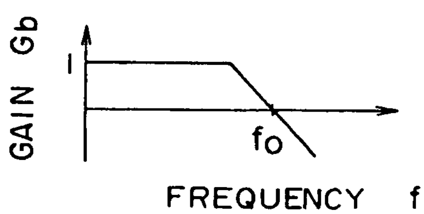


FIG. 14

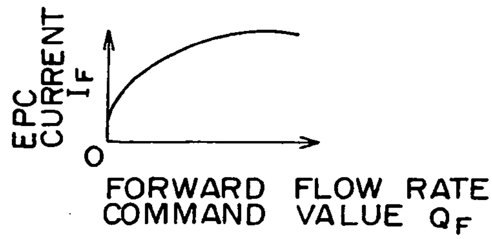


FIG. 15

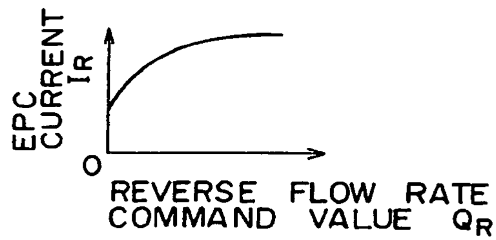


FIG. 16

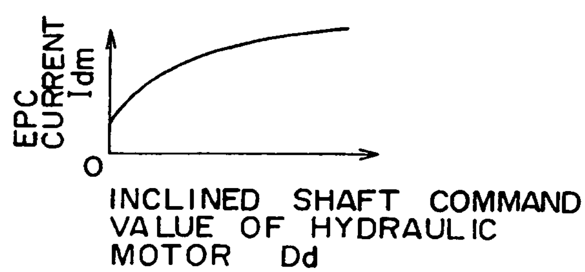


FIG. 17

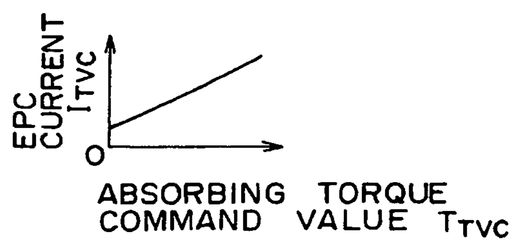


FIG. 18

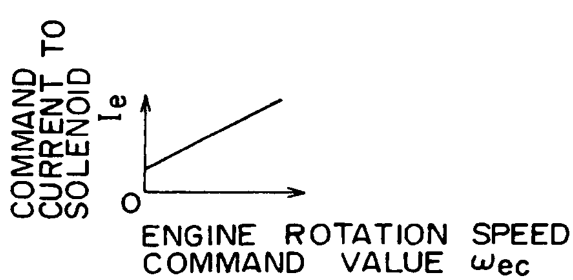


FIG. 19

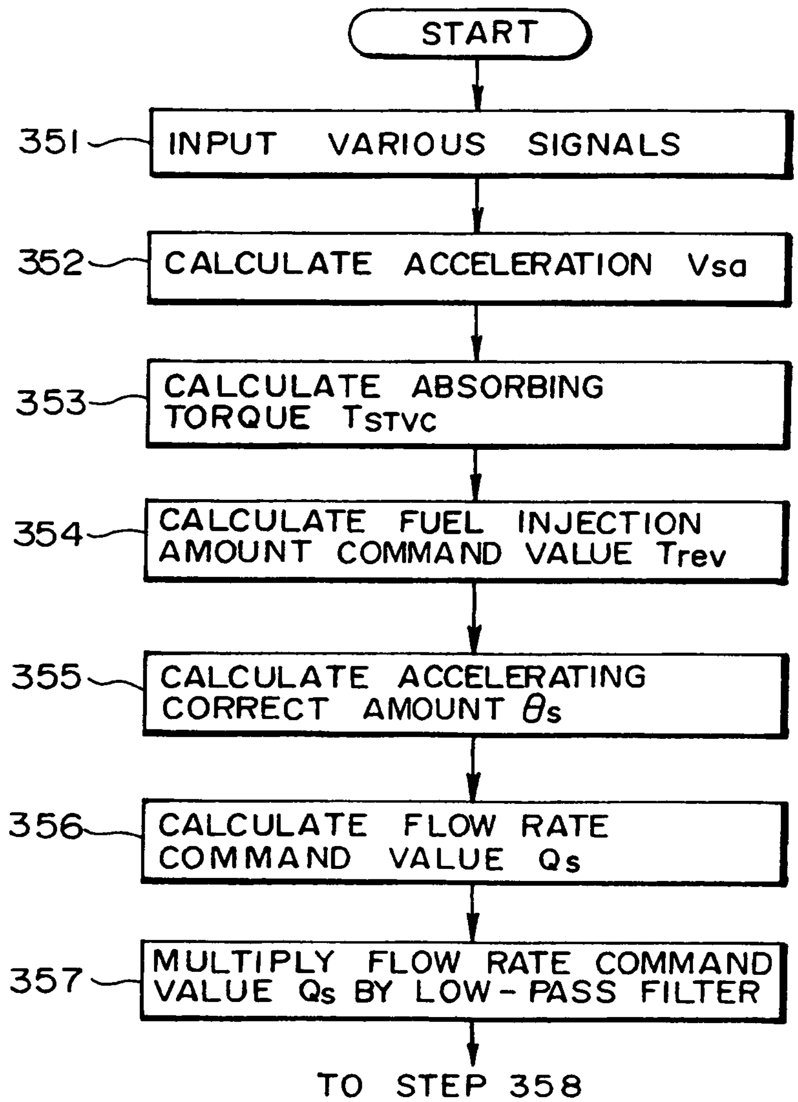

START
351
INPUT VARIOUS SIGNALS
352
CALCULATE ACCELERATION Vsa
353
CALCULATE ABSORBING TORQUE TSTVC
354
CALCULATE FUEL INJECTION AMOUNT COMMAND VALUE Trev
355
CALCULATE ACCELERATING CORRECT AMOUNT θs
356
CALCULATE FLOW RATE COMMAND VALUE Qs
357
MULTIPLY FLOW RATE COMMAND VALUE Qs BY LOW-PASS FILTER
TO STEP 358

FIG. 20

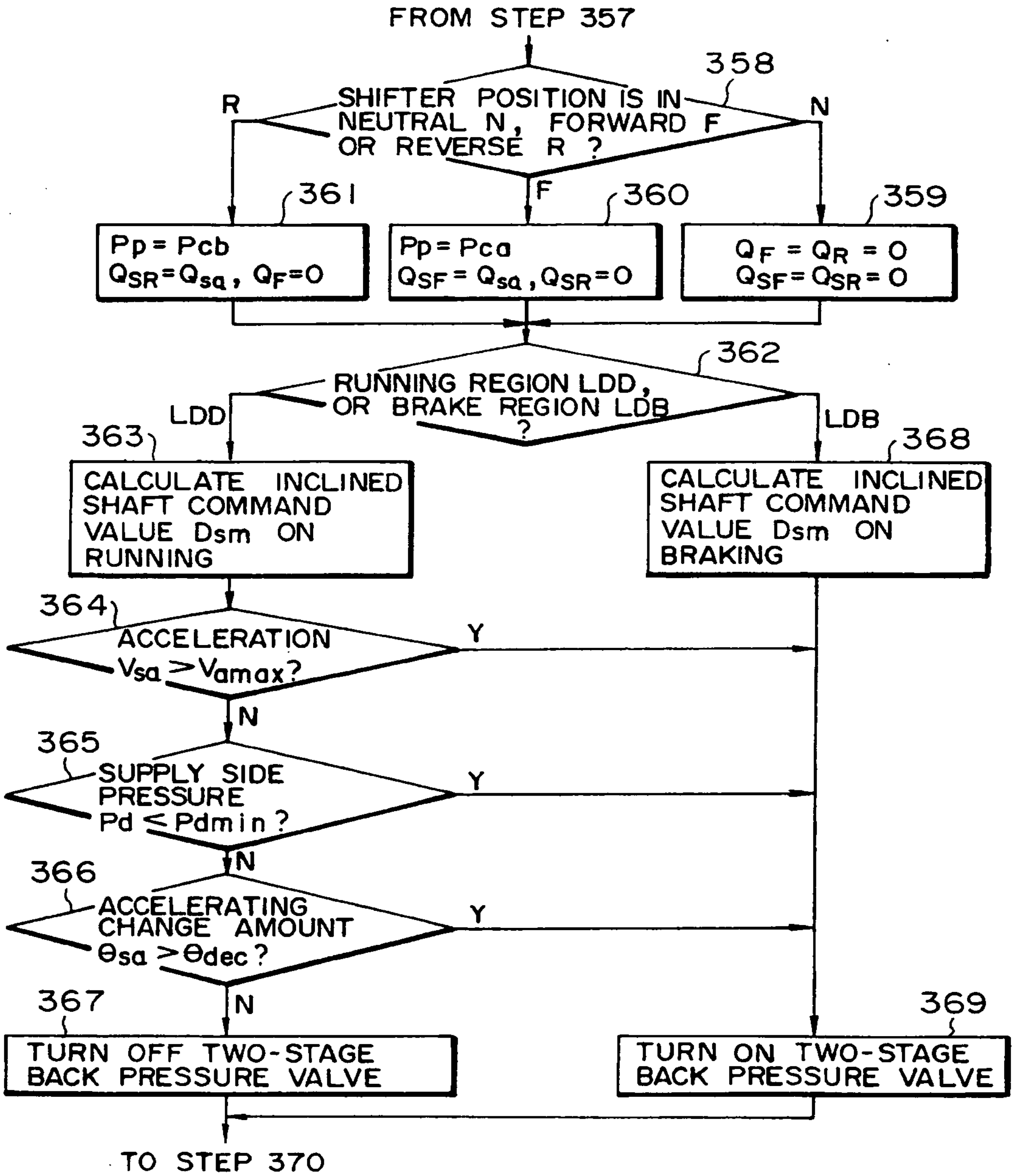

FROM STEP 357
358
SHIFTER POSITION IS IN NEUTRAL N, FORWARD F OR REVERSE R ?
R
N
F
361
Pp = Pcb
$Q_{SR}=Q_{sa}$, $Q_F=0$
360
Pp = Pca
$Q_{SF}=Q_{sa}$, $Q_{SR}=0$
359
$Q_F = Q_R = 0$
$Q_{SF}=Q_{SR}=0$
362
RUNNING REGION LDD, OR BRAKE REGION LDB ?
LDD
LDB
363
CALCULATE INCLINED SHAFT COMMAND VALUE Dsm ON RUNNING
368
CALCULATE INCLINED SHAFT COMMAND VALUE Dsm ON BRAKING
364
ACCELERATION $V_{sa} > V_{amax}$?
Y
N
365
SUPPLY SIDE PRESSURE $Pd < Pdmin$?
Y
N
366
ACCELERATING CHANGE AMOUNT $\theta_{sa} > \theta_{dec}$?
Y
N
367
TURN OFF TWO-STAGE BACK PRESSURE VALVE
369
TURN ON TWO-STAGE BACK PRESSURE VALVE
TO STEP 370

FIG.21

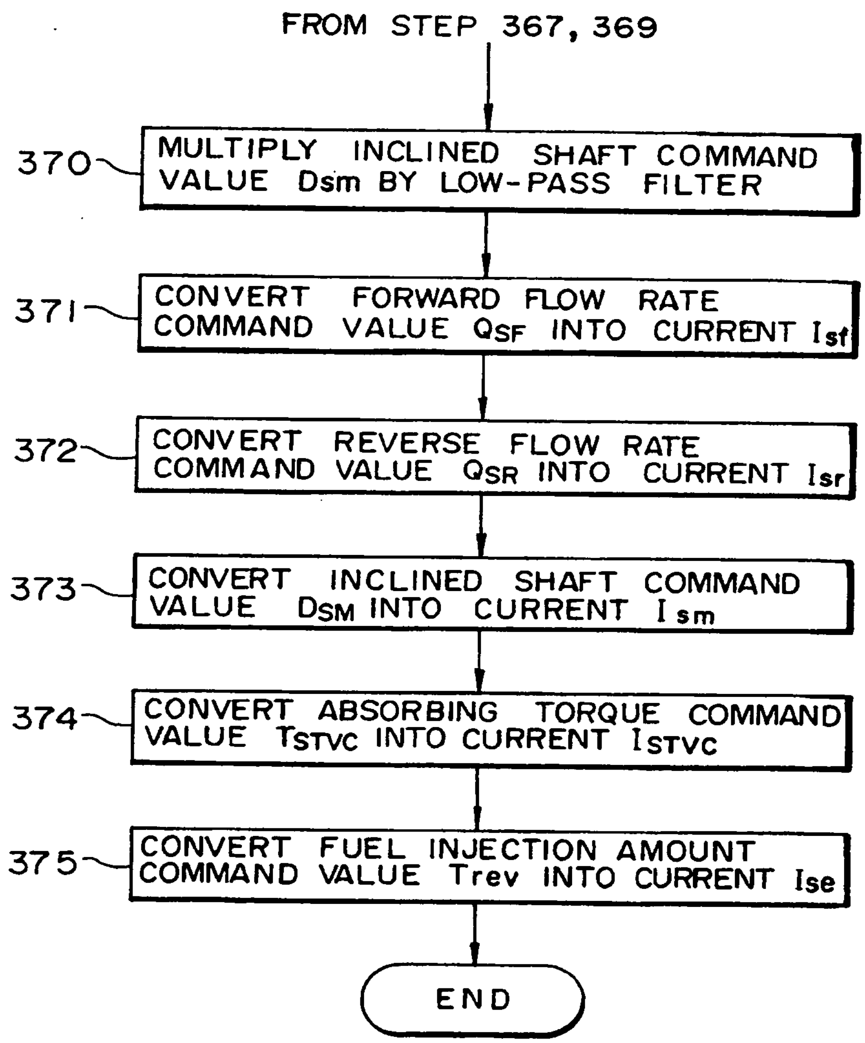

FROM STEP 367, 369
370
MULTIPLY INCLINED SHAFT COMMAND VALUE Dsm BY LOW-PASS FILTER
371
CONVERT FORWARD FLOW RATE COMMAND VALUE QSF INTO CURRENT Isf
372
CONVERT REVERSE FLOW RATE COMMAND VALUE QSR INTO CURRENT Isr
373
CONVERT INCLINED SHAFT COMMAND VALUE DSM INTO CURRENT Ism
374
CONVERT ABSORBING TORQUE COMMAND VALUE TSTVC INTO CURRENT ISTVC
375
CONVERT FUEL INJECTION AMOUNT COMMAND VALUE Trev INTO CURRENT Ise
END

FIG. 22

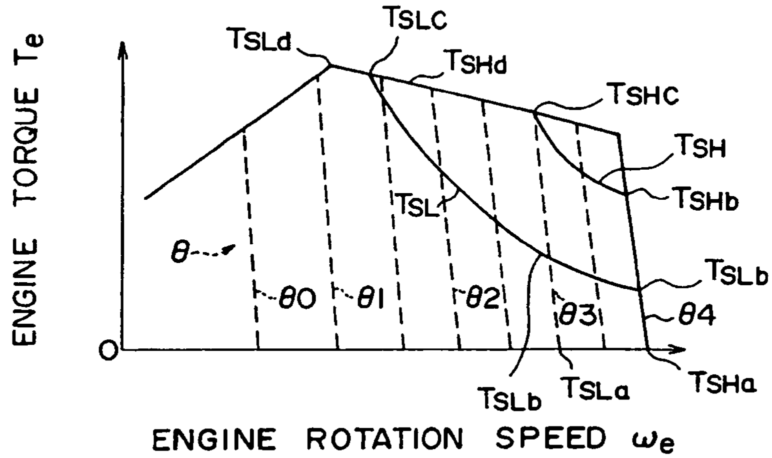


FIG. 23

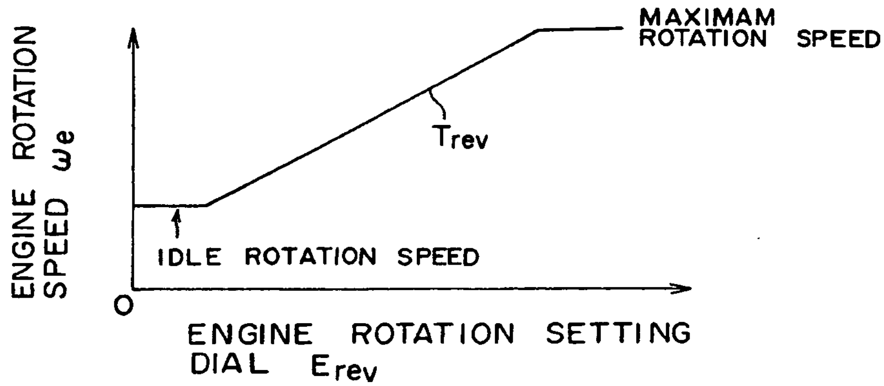


FIG. 24

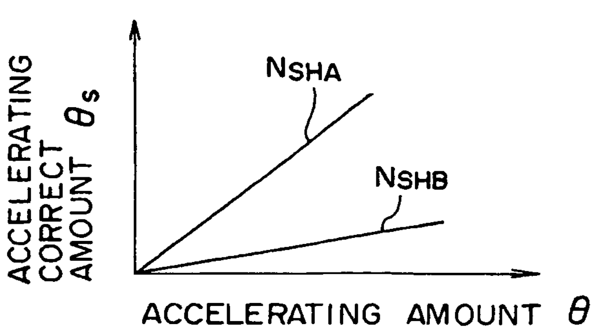


FIG. 25

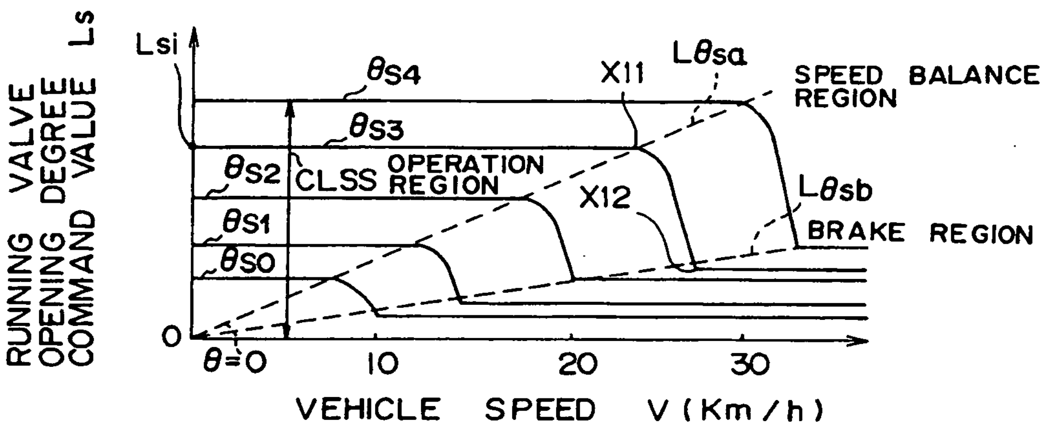


FIG. 26

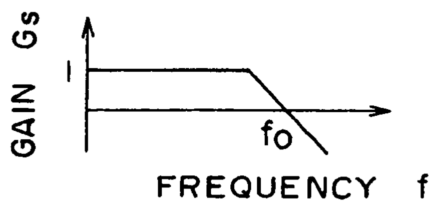


FIG. 27

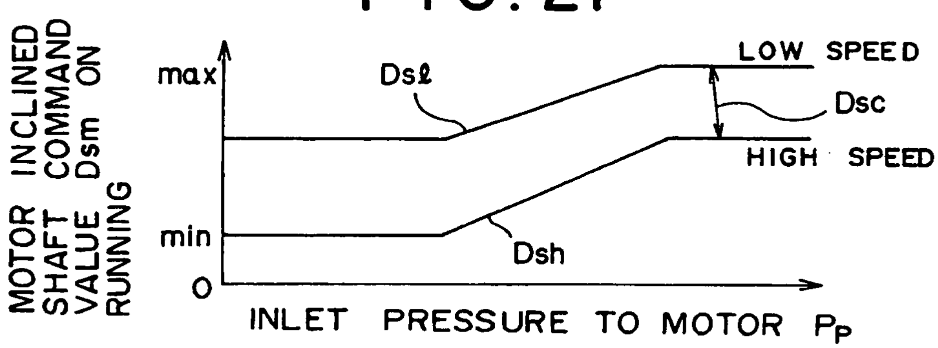


FIG. 28

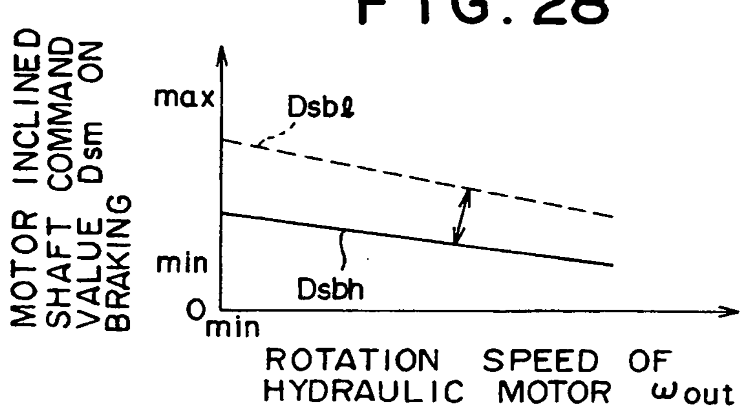


FIG.29

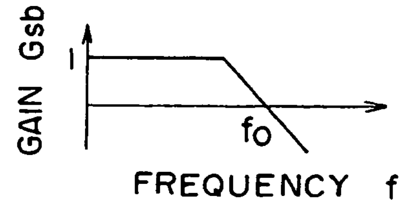


FIG.30

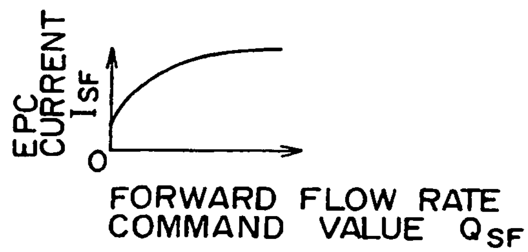


FIG.31

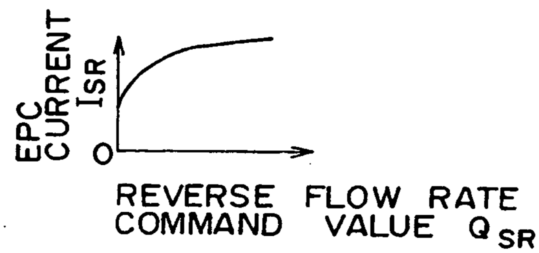


FIG.32

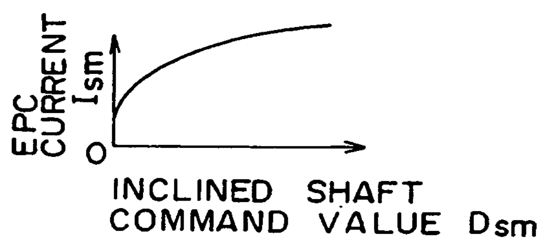


FIG.33

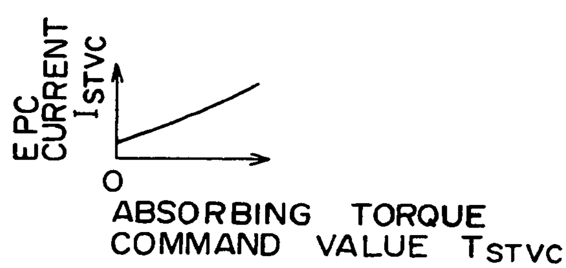


FIG.34

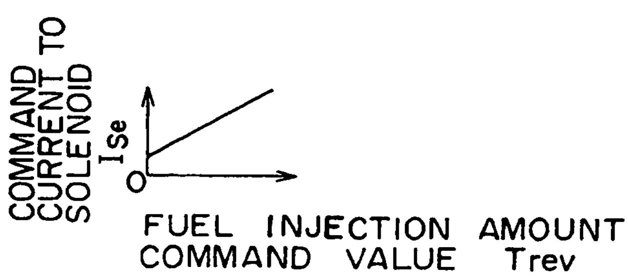


FIG. 35

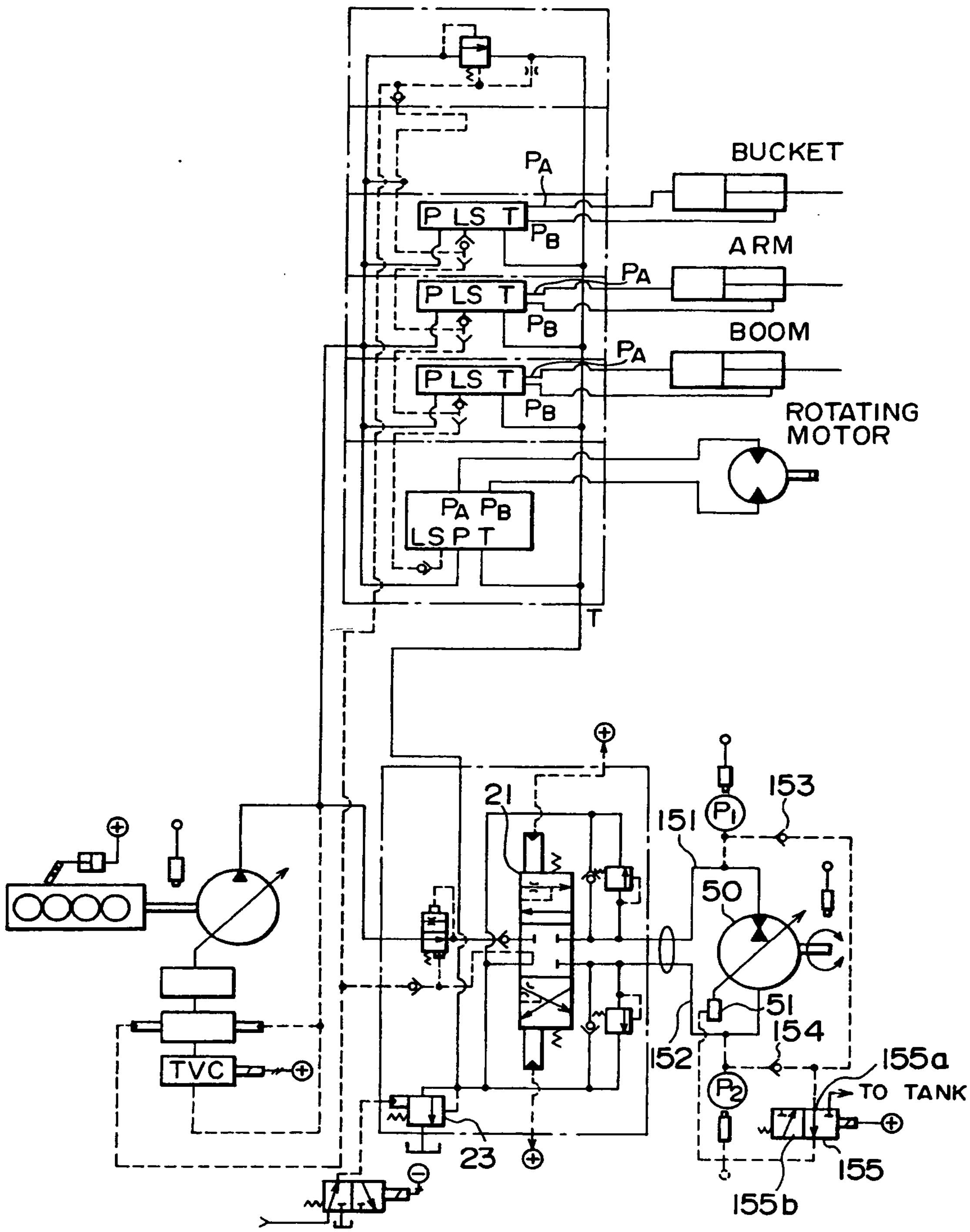

BUCKET
ARM
BOOM
ROTATING MOTOR
P LS T
PA
PB
PA PB
LS P T
T
21
23
TVC
50
51
151
152
153
154
155
155a
155b
TO TANK
P1
P2

FIG. 36

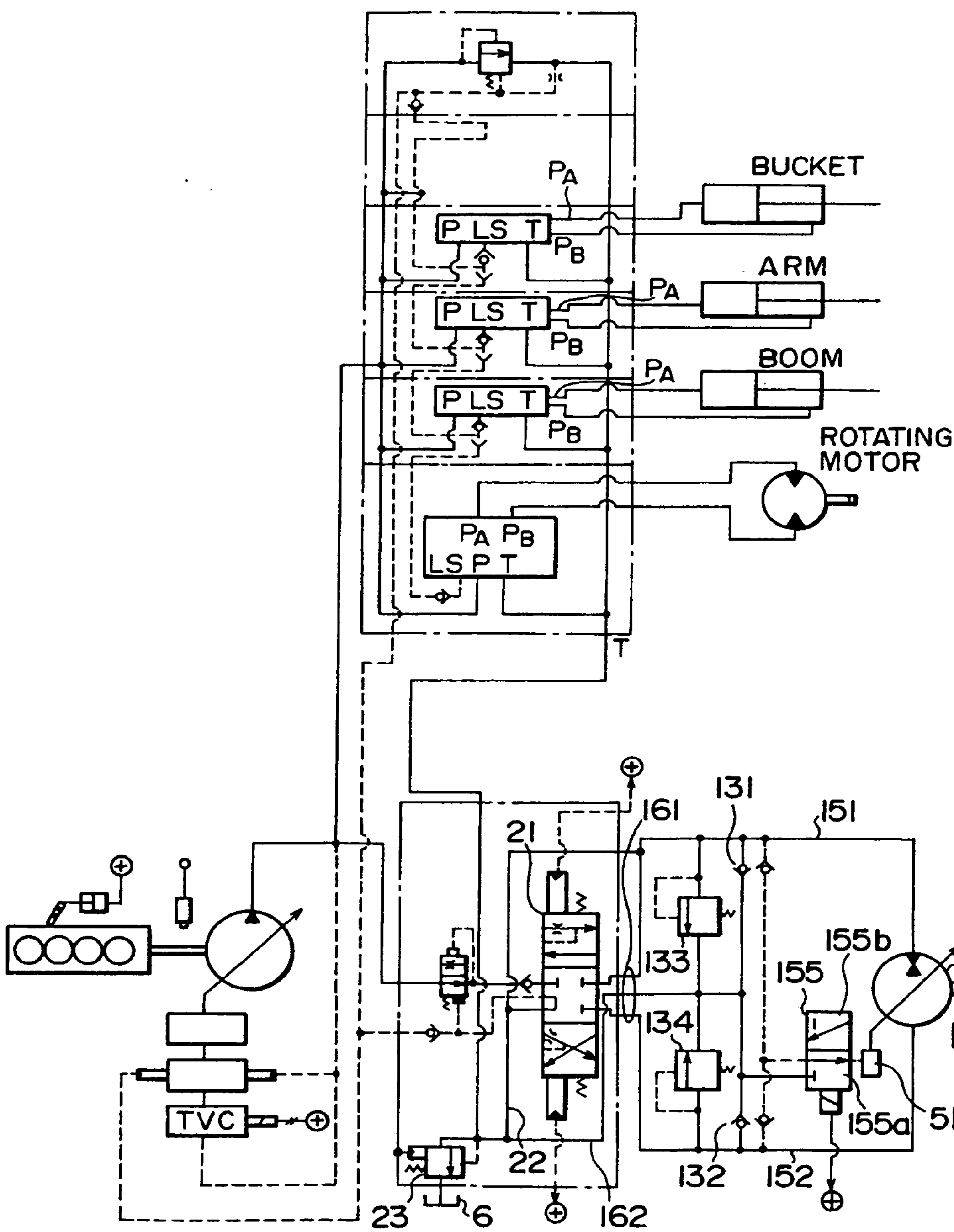
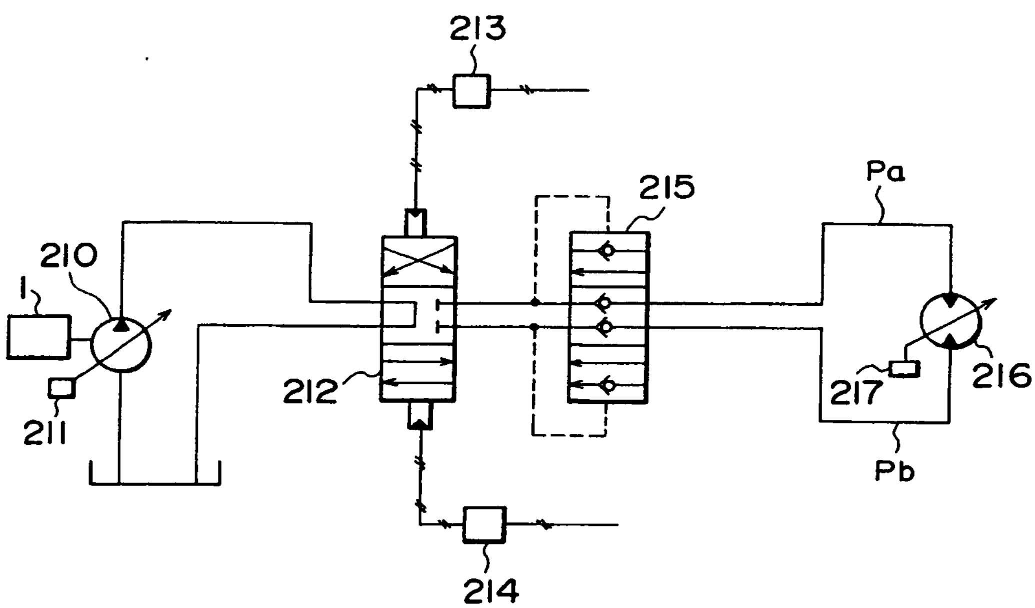

PA
BUCKET
P LS T
PB
ARM
PA
P LS T
PB
BOOM
PA
P LS T
PB
ROTATING
MOTOR
PA PB
LS P T
T
161
131
151
21
155b
50
133
155
134
TVC
22
155a
51
23
6
162
132
152

FIG. 37
PRIOR ART



INTERNATIONAL SEARCH REPORT

International application No. PCT/JP95/02279

A. CLASSIFICATION OF SUBJECT MATTER

Int. Cl6 F16H61/40, F15B11/02

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)

Int. Cl6 F16H61/40, F15B11/02

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practicable, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	JP, 60-88730, A (Sumitomo Heavy Industries, Ltd.), May 18, 1985 (18. 05. 85)	1-4, 7-10 12, 13
Y	JP, 58-193961, A (Toyota Motor Corp.), November 11, 1983 (11. 11. 83)	1-4, 7-10 12, 13
	JP, 3-229003, A (Hitachi Construction Machinery Co., Ltd.),	
Y	October 11, 1991 (11. 10. 91)	4
A		11

☐ Further documents are listed in the continuation of Box C. ☐ See patent family annex.

* Special categories of cited documents:

"A" document defining the general state of the art which is not considered to be of particular relevance

"E" earlier document but published on or after the international filing date

"L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)

"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
January 31, 1996 (31. 01. 96)	February 27, 1996 (27. 02. 96)
Name and mailing address of the ISA/ Japanese Patent Office	Authorized officer
Facsimile No.	Telephone No.

Form PCT/ISA/210 (second sheet) (July 1992)

INTERNATIONAL SEARCH REPORT

International application No.
PCT/JP95/02279

Box I Observations where certain claims were found unsearchable (Continuation of item 1 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. [X] Claims Nos.: 6
because they relate to subject matter not required to be searched by this Authority, namely:
A method of calculating absorbed torque is a scientific theory.

2. [5] Claims Nos.: 5
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:
Claims are made indefinite by the description "controlling within a predetermined range".

3. [ ] Claims Nos.:
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box II Observations where unity of invention is lacking (Continuation of item 2 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

1. [ ] As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.

2. [ ] As all searchable claims could be searched without effort justifying an additional fee, this Authority did not invite payment of any additional fee.

3. [ ] As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:

4. [ ] No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest
[ ] The additional search fees were accompanied by the applicant's protest.
[ ] No protest accompanied the payment of additional search fees.

Form PCT/ISA/210 (continuation of first sheet (1)) (July 1992)